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ENERGY CONVERSION ALTERNATIVES STUDY
-ECAS-
GENERAL ELECTRIC PHASE I FINAL REPORT

VOLUME II, ADVANCED ENERGY CONVERSION SYSTEMS
Part 1, Open-Cycle Gas Turbines

by

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16. Abstract

A parametric study was performed to assist in the development of a data base for the comparison of advanced energy conversion systems for utility applications using coal or coal-derived fuels. Estimates of power plant performance (efficiency), capital cost, cost of electricity, natural resource requirements, and environmental intrusion characteristics were made for ten advanced conversion systems. Over 300 parametric points were analyzed to estimate the potential of these systems. Emphasis of the study was on the energy conversion system in the context of a base loaded utility power plant. Although cases employing transported coal-derived fuels were included in the study, the fuel processing step of converting coal to clean fuels was not investigated except for cases where a low-Btu gasifier was integrated with the power plant. All power plant concepts were premised on meeting emission standards requirements. The investigative approach focused on achieving consistency and comparability in the analysis of the various conversion systems. Recognized advocate organizations were employed to analyze their respective cycles and to present their analyses for power plant integration by the GE systems evaluation team. Wherever possible, common subsystems and components for the various systems were treated on a uniform basis. A steam power plant (3500 psig, 1000 F, 1000 F) with a conventional coal-burning furnace-boiler was analyzed as a basis for comparison. Combined cycle gas/steam turbine system results indicated competitive efficiency and a lower cost of electricity compared to the reference steam plant. The Open-Cycle MHD system results indicated the potential for significantly higher efficiency than the reference steam plant but with a higher cost of electricity. The information contained in this report constitutes results from the first phase of a two phase effort. In Phase II, a limited number of concepts will be investigated in more detail through preparation of conceptual designs and an implementation assessment including preparation of R&D plans estimating the resources and time required to bring the systems to commercial fruition.

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FOREWORD

The work described in this report is a part of the Energy Conversion Alternatives Study (ECAS)—a cooperative effort of the Energy Research and Development Administration, the National Science Foundation, and the National Aeronautics and Space Administration.

This General Electric contractor report for ECAS Phase I is contained in three volumes:

Volume I - Executive Summary

Volume II - Advanced Energy Conversion Systems

- Part 1 - Open-Cycle Gas Turbines
- Part 2 - Closed Turbine Cycles
- Part 3 - Direct Energy Conversion Cycles

Volume III - Energy Conversion and Subsystems and Components

- Part 1 - Bottoming Cycles and Materials of Construction
- Part 2 - Primary Heat Input Systems and Heat Exchangers
- Part 3 - Gasification, Process Fuels, and Balance of Plant

In addition to the principal authors listed, members of the technical staffs of the following subcontractor organizations developed information for the Phase I data base:

General Electric Company
- Advanced Energy Programs/Space Systems Department
- Direct Energy Conversion Programs
- Electric Utility Systems Engineering Department
- Gas Turbine Division
- Large Steam Turbine-Generator Department
- Medium Steam Turbine Department
- Projects Engineering Operation/I&SE Engineering Operation
- Space Sciences Laboratory

Actron, a Division of McDonnell Douglas Corporation

Argonne National Laboratory

Avco Everett Research Laboratory, Incorporated

Bechtel Corporation

Foster Wheeler Energy Corporation

Thermo Electron Corporation

This General Electric contractor report is one of a series of three reports discussing ECAS Phase I results. The other two reports are the following: Energy Conversion Alternatives Study (ECAS), Westinghouse Phase I Final Report (NASA CR-134941), and NASA Report (NASA TMX-71855).

TABLE OF CONTENTS

TABLE OF CONTENTS (Cont'd)

LIST OF ILLUSTRATIONS

Volume II, Part 1

LIST OF TABLES

LIST OF TABLES (Cont'd)

Summary

ADVANCED ENERGY CONVERSION SYSTEMS

The objective of Phase I of the Energy Conversion Alternatives Study (ECAS) for coal or coal-derived fuels was to develop a technical-economic information base on the ten energy conversion systems specified for investigation. Over 300 parametric variations were studied in an attempt to identify system and cycle conditions which indicate the best potential of the energy conversion concept. This information base provided a foundation for selection of energy conversion systems for more in-depth investigation in the conceptual design portion of the ECAS study. The systems for continued study were specified by the ECAS Interagency Steering Committee.

The technical-economic results include efficiency, capital cost and cost of electricity. For reference purposes a steam cycle (3500 psi/1000 F/1000 F [2.41×10^7 N/m^2/811 K/811 K]) with conventional coal burning furnace, stack gas cleanup and wet mechanical draft cooling towers was analyzed with the same analysis procedure employed for the advanced systems. This reference steam plant had an efficiency of approximately 37 percent. The open-cycle MHD system was the only plant to show efficiencies approaching 50 percent. A group of cycles-advanced steam, supercritical CO_2, liquid metal topping, and inert gas MHD— were estimated to have efficiencies in the 40 to 45 percent range.

The energy conversion systems with capital costs significantly lower than the reference steam plant were those with short construction times and simple construction, i.e., open-cycle gas turbines and low-temperature fuel cells. The more complex plants, i.e., open- and closed-cycle MHD and liquid metal topping, required longer construction time and were higher in capital cost.

Efficiency and capital cost are a part of the total technical-economic evaluation. The combination of these characteristics with the cost of fuel and operation and maintenance costs results in a cost of electricity for more complete comparisons. The only systems which were consistently lower than the reference steam plant's 30 mills/kWh at 65 percent capacity factor were the open-cycle gas turbine-combined cycle. MHD, supercritical CO_2, liquid metal top topping, and high-temperature fuel cells had a higher cost of electricity than the reference steam plant, as did many of the advanced steam cases because of their higher capital costs. The low capital cost plants— (low-temperature fuel cells and open cycle gas turbine, recuperative) utilized clean fuels and consequently had high fuel charges. These systems would be more economically applicable to peaking or mid-range duty.

ADVANCED ENERGY CONVERSION SYSTEMS

Many advanced energy conversion techniques which can use coal or coal-derived fuels have been advocated for power generation applications. Conversion systems advocated have included open- and closed-cycle gas turbine systems (including combined gas turbine-steam turbine systems), supercritical CO_2 cycle, liquid metal Rankine topping cycles, magnetohydrodynamics (MHD), and fuel cells. Advances have also been proposed for the steam systems which now form the backbone of our electric power industry. These advances include the use of new furnace concepts and higher steam turbine inlet temperatures and pressures. Integration of a power conversion system with a coal processing plant producing a clean low-Btu gas for use in the power plant is still another approach advocated for energy conserving, economical production of electric power. Studies of all these energy conversion techniques have been performed in the past. However, new studies performed on a common basis and in light of new national goals and current conditions are required to permit an assessment of the relative merits of these techniques and potential benefits to the nation.

The purpose of this contract is to assist in the development of an information base necessary for an assessment of various advanced energy conversion systems and for definition of the research and development required to bring these systems to fruition. Estimates of the performance, economics, natural resource requirements and environmental intrusion characteristics of these systems are being made on as comparable and consistent a basis as possible leading to an assessment of the commercial acceptability of the conversion systems and the research and development required to bring the systems to commercial reality. This is being accomplished in the following tasks:

Task I	Parametric Analysis	(Phase I)
Task II	Conceptual Designs	(Phase II)
Task III	Implementation Assessment	(Phase II)

This investigation is being conducted under the Energy Conversion Alternatives Study (ECAS) under the sponsorship of Energy Research and Development Administration (ERDA), National Science Foundation (NSF), and National Aeronautics and Space Administration (NASA). The control of the program is under the direction of an Interagency Steering Committee with participation of the supporting agencies. The NASA Lewis Research Center is responsible for project management of this study.

The information presented in this report describes the results produced in the Task I portion of this study. The emphasis

in this task was placed upon developing an information base upon which comparisons of Advanced Energy Conversion Techniques using coal or coal-derived fuels can be made. The Task I portion of the study was directed at a parametric variation of the ten advanced energy conversion systems under investigation. The wide-ranging parametric study was performed in order to provide data for selection by the Interagency Steering Committee of the systems and specific configurations most appropriate for Task II and III studies.

The Task II effort will involve a more detailed evaluation of seven advanced energy conversion systems and result in a conceptual design of the major components and power plant layout. The Task III effort will produce the research and development plans which would be necessary to bring each of the seven Task II systems to a state of commercial reality and then to assess their potential for commercial acceptability.

A prime objective of this study was to produce results which had a cycle-to-cycle consistency. In order to accomplish this objective and still ensure that each system was properly advocated, an organization which is or had been a proponent of the prime cycle was selected to advocate the energy conversion system and to analyze the performance and economics of the prime cycle portion of the energy conversion system, i.e., the parts of the system which were novel or unique to the system. The remaining subsystems, e.g., fuel processing, furnaces, bottoming cycles, balance of plant, were analyzed by technology specialist organizations which presently have responsibility for supplying these subsystems for utility applications. The final plant configuration and performance were produced by the General Electric Corporate Research and Development study team and this group performed the critical integration of the final plant concept. This methodology was used to provide a system-to-system consistency while maintaining the influence of a cycle advocate.

The ten energy conversion systems under investigation in this study are defined and analyzed in this volume of the report. These include:

1. Open-cycle Gas Turbine Recuperative

   - with clean and semi-clean fuels produced from coal
   - with and without organic bottoming cycles

2. Open-Cycle Gas Turbine

   - with air and water cooling of the gas turbine hot gas path
   - with clean and semi-clean fuels from coal and integrated low-Btu gasifiers

3. **Closed-Cycle Gas Turbine**

   - with helium working fluid
   - with a variety of direct coal and clean fuel furnaces
   - with and without organic and steam bottoming cycles

4. Supercritical CO_2 Cycle

   - with basic and recompression cycle variations
   - with a variety of direct coal and clean coal-derived fuel furnaces

5. Advanced Steam Cycle

   - with both throttle and/or reheat temperatures greater than present practice (1000 F [811 K])
   - with a variety of direct coal and clean coal-derived fuel furnaces

6. Liquid Metal Topping Cycle

   - with potassium and cesium as working fluids
   - with a variety of direct coal and clean fuel furnaces

7. Open-Cycle MHD

   - with direct coal and semi-clean fuel combustion
   - with standard steam and gas turbine bottoming

8. Closed-Cycle Inert Gas MHD

   - with parallel and topping configurations
   - with both direct coal and semi-clean fuel utilization

9. Closed-Cycle Liquid Metal MHD

   - with mixture of liquid sodium and helium as working fluids
   - with standard steam bottoming
   - with a variety of direct coal and clean fuel furnaces

10. Fuel Cells

   - both high and low temperature (less than 300 F [422 K])
   - with employment of clean process fuels for low temperature cells and low-Btu gasification at high temperature cells

The subsystems which complete the energy conversion system are discussed in Volume III of this report. The results as presented in the following sections include the total energy conversion system.

Section 1

ANALYTICAL APPROACH

The primary objective of the Task I Study was to perform a comparative evaluation study of Advanced Energy Conversion Systems and to investigate variations of cycle parameters and configurations for each system. The emphasis of the study was placed upon obtaining a consistency of analysis so that the conversion systems can be compared on an equal relative basis. The analytical approach was adopted to accomplish these objectives.

GROUNDRULES FOR STUDY

The focus of this report was on baseloaded central power plants for electrical generation. The rating of each plant was based on an average daily ambient temperature at a "Middletown, U.S.A." site. These conditions were 59 F (288 K) dry bulb temperature and 60 percent relative humidity. Insofar as possible, the generator and coal handling equipment were costed to handle the maximum power output which could be expected at temperatures to 20 F (266 K). The heat rejection equipment was sized and costed to meet the nominal plant output during the average dry conditions and still maintain load for the 5 percent summer condition, 94 F (307 K).

A range of plant ratings from 24 MW to 2400 MW was evaluated. This variation was utilized with fuel cells and simple cycle gas turbines at the low end of the range and liquid metal topping and open-cycle MHD at the top of the range.

The design objective for the plants was a 30-year lifetime and a common goal of 90 percent availability.

The economic evaluations were made with a 65 percent capacity factor. An 18 percent fixed charge on capital costs was used. All costs were evaluated in mid-1974 dollars. Interest and escalation were estimated during construction. These rates were 6 1/2 percent escalation and a 10 percent interest charge, both compounded. The cash flow during plant construction was assumed to be an "S" function similar to that indicated in Reference 1.

The fuels which were utilized in this study are coal or coal-derived fuels all utilized by the power plant in ways which would meet the specified environmental criteria. The three approaches to processing this fuel are: 1) direct combustion with cleanup procedures employed during combustion or on the exhaust gas, 2) integrated low-Btu gasification, and 3) transportable clean fuels. The latter fuel was priced on the basis of a fuel delivered at the fence of the plant for a specified price.

The heat rejection systems were selected to minimize thermal pollution. Wet, mechanical draft, cooling towers were utilized as the primary heat rejection scheme. However, each cycle was also evaluated with dry cooling towers.

The efficiencies and heat rate of the energy conversion systems were all based upon the higher heating value (HHV) of the fuel.

A more complete specification of the study groundrules is given in Appendix A.

ENERGY CONVERSION SYSTEM VARIATION

In order to facilitate this investigation, the energy conversion system was divided into three parts: 1) Heat Input System, 2) Cycle, and 3) Heat Rejection System.

Heat Input System

In open-cycle systems, the combustion gas is the cycle working fluid. In this concept, the combustion system was evaluated as a part of the conversion cycle. In closed-cycle systems, the thermal input to the cycle was through a heat exchanger device. The input heat exchanger and combustor for these systems were handled separately from the prime cycle (system for conversion of thermal to electrical energy).

Cycles

Ten specific energy conversion systems were evaluated:

- Open-Cycle Gas Turbine-Simple and Recuperative
- Open-Cycle Gas Turbine-Combined Cycle
- Closed-Cycle Gas Turbine
- Supercritical CO_2 Cycle
- Advanced Steam Cycle
- Liquid Metal Topping Cycle
- Open-Cycle MHD
- Closed-Cycle Inert Gas MHD
- Closed-Cycle Liquid Metal MHD
- Fuel Cells

Many of the prime or topping cycles require the cascading of energy from the prime cycle into a bottoming cycle in order to make the overall concept viable. Both steam and organic fluid cycles were evaluated for coupling with the prime cycle. To the maximum degree possible, these bottoming cycles were selected to be state-of-the-art equipment. This permitted a greater focus to be placed on the prime cycle perturbations and evaluations.

Heat Rejection Systems

The heat rejection from the prime and/or bottoming cycle was accomplished in an environmentally acceptable manner. The

emphasis of this study was placed on wet and dry cooling towers as heat rejection devices. The coupling of the cooling tower with the cycle heat rejection-heat exchanger, i.e., precooler or condenser, was by a water loop. This water loop was open for wet cooling towers and closed for dry cooling towers.

ANALYSIS PHILOSOPHY

The emphasis of the Task I Study was to explore a wide range of parameters and support subsystems combinations which could make the prime cycle attractive for a utility application. In order to ensure that each energy conversion system was evaluated and projected in its best light, an advocate organization was subcontracted to analyze each prime cycleportion of the system. These organizations were well suited for this analysis since they either are or have been proponents of the cycle configuration which they evaluated. Because of the short time available for Task I, parametric cases had to be selected at the start of the program based on the collective insight and judgment of the contractor team, including the system advocate, and the government. Little opportunity existed for truly optimizing the systems.

The subsystems which support the prime cycles were evaluated in common for each cycle. The evaluation was performed by the same group working with a common set of assumptions, design philosophy, and costing information to help ensure a cycle-to-cycle uniformity in the overall analysis. The components which were evaluated in common for each system were:

- Fuel processing
- Primary heat exchangers (furnaces)
- Bottoming cycles
- Heat exchangers
- Heat rejection systems
- Balance-of-plant considerations

The organizational responsibility for these analyses is shown in Table 1-1. This table demonstrates the across-the-board evaluation of common subsystems. The advocate organization for each prime cycle is also indicated on this table.

The objective of the Task I Study was to obtain parametric information as critical parameters and subsystems were varied for each energy conversion system. In order to facilitate the evaluation procedure, one or more base cases were selected for each energy conversion system. The parametric variations were the evaluated as perturbations from these base cases. The information which was developed for each base case and parametric point consists of overall efficiency, plant capital cost, and cost of electricity. In addition for each base case, the following information was generated; details of major components (e.g., size, weight, cost), natural resources required, and environmental intrusion.

Table 1-1

Subsystem Responsibility

Subsystem \ System Advocate Organization	Open Cycle Gas Turbine: Regenerative GE/GT	Open Cycle Gas Turbine: Combined GE/GT	Closed Cycle Gas Turbine GE/GT	Supercritical CO_2 Actron	Advanced Steam GE/LS	Metal Vapor Topping GE/SD	MHD: Open Avco	MHD: Closed Inert Gas GE/SD	MHD: Closed Liquid Metal ANL	Fuel Cells GE/DECP
Heat input										
Clean fuel production	GE/Gasif	GE/Gasif	GE/Gasif	GE/Gasif	GE/Gasif	GE/Gasif	GE/Gasif	GE/Gasif	GE/Gasif	GE/Gasif
Combustion and primary heat exchangers	--	--	FW	FW	FW	FW	--	FW	FW	--
Pressurizing gas turbine	--	--	GE/GT	GE/GT	GE/GT	GE/GT	--	--	GE/GT	--
Prime cycle										
Rotational equipment	Advocate	Advocate	Advocate	Advocate/GE	Advocate	Advocate	--	-	--	--
Combustor	Advocate	Advocate	--	--	--	--	Advocate	--	--	--
Direct energy conversion	--	--	--	--	--	--	Advocate	Advocate	Advocate	Advocate
Inverters	--	--	--	--	--	--	GE	GE	GE	GE
Regenerators	Advocate	--	Advocate/GE HT	Advocate/GE HT	--	--	--	Advocate	--	--
Seed addition	--	--	--	--	--	--	Advocate	Advocate	Advocate	--
Heat recovery heat exchangers										
Fluid boiler	GE/HT, TE	GE/MS	GE/MS/HT, TE	--	--	FW	FW	FW	FW	FW
Combustion air present	--	--	--	-	--	--	Advocate/FW	--	--	--
Bottoming cycle										
Rotational equipment	TE	GE/MS	GE/MS, TE	--	--	GE/LS	GE/LS/GT	GE/LS	GE/LS/GT	GE/LS
Heat rejection										
Condensers	GE/HT, TE	Bechtel	GE/HT, Bechtel, TE	—	Bechtel	Bechtel	Bechtel	Bechtel	Bechtel	Bechtel
Precoolers	--	--	GE/HT	GE/HT	--	--	--	GE/HT	GE/HT	--
Cooling towers	Bechtel	Bechtel	Bechtel	Bechtel	Bechtel	Bechtel	Bechtel	Bechtel	Bechtel	Bechtel
Environmental intrusion										
Emission characteristics	Advocate	Advocate	FW	FW	FW	FW	Advocate, FW	FW	FW	FW
Emission control equipment	Advocate	Advocate, GE/Gasif	GE/Gasif, FW	GE/Gasif, FW	Bechtel GE/Gasif, FW	GE/Gasif, FW	Bechtel, Advocate	Bechtel	GE/Gasif, FW	GE/Gasif
System integration	GE	GE	GE	GE	GE	GE	GE	GE	GE	GE
Balance of plant	Bechtel	Bechtel	Bechtel	Bechtel	Bechtel	Bechtel	Bechtel	Bechtel	Bechtel	Bechtel

ANL = Argonne National Laboratory
FW = Fcster-Wheeler Corporation
GE/DECP = GE Direct Energy Conversion Program
GE/Gasif = GE Gasification Project
GE/GT = GE Gas Turbine Products Division
GE/HT = GE Heat Transfer Products
GE/MS = GE Medium Steam Turbine-Generator Products Department
GE/LS = GE Turbine Department
GE/SD = GE Space Division
TE = Thermo Electron Corporation

In these studies, three different efficiences were calculated:

1. Thermodynamic efficiency

$$\frac{\text{Total of gross electrical energy generated by prime cycle and bottoming cycle excluding generation by pressurizing gas turbines}}{\text{Thermal energy into the prime cycle}}$$

2. Power plant efficiency

$$\frac{\text{Net electrical energy generated}}{\text{Higher heating value of power plant fuel}}$$

3. Overall efficiency (coal pile to bus bar)

(Power plant efficiency) x (Process fuel conversion efficiency)

STUDY PARAMETERS

Specific coals were specified for analysis in this study. These coals are representative of the major coal reserves in the United States. Their characteristics are shown in Table 1-2.

Table 1-2

COAL SPECIFICATIONS*

	Illinois No. 6	Montana Sub-bituminous	North Dakota Lignite
Higher heating value (Btu/lb)	10,788	8944	6890
Proximate analysis			
Moisture	13	24.3	36.7
Volatile	36.7	28.6	26.6
Fixed carbon	40.7	39.6	30.5
Ash	9.6	7.5	6.2
Ultimate analysis (partial list)			
Sulfur	3.9	0.8	0.7
Nitrogen	1.0	0.8	0.6
Cost (delivered) ($/MM Btu)	0.85	0.85	0.85

*Coal specifications supplied

Each of these coals was used as an energy source by the various energy conversion systems. As previously stated, the coal was utilized at the site in either direct combustion or low-Btu gasification. These coals could also be employed at off-site locations to produce a clean fuel. The clean fuels and their characteristics are shown in Table 1-3. The variety of processes were selected to allow the energy conversion concepts to be analyzed with synthetic fuels which simulate: 1) natural gas, high-Btu gas, 2) distillate or clean oil (char oil energy development) [COED]), and 3) a residual or semi-clean oil (solvent refined coal [SRC]). The delivered cost of the clean fuels was specified by NASA.

Table 1-3

CLEAN FUEL SPECIFICATION

Fuel Parameters	High-Btu Gas	Intermediate Btu Gas	Low-Btu (free standing)	Hydrogen	Solvent Refined Coal	COED
Higher heating value (Btu/lb)	22,674	6350	2535	54,047	15,682	17,041
Cost delivered ($/MM Btu)	2.60	2.00	2.08	2.50	1.80	2.60
Conversion efficiency (percent)	50	70	68	61	78	56

For each closed cycle, at least two direct coal combustion schemes were employed. Both of these direct coal combustion concepts feature sulfur capture during fluid bed combustion and both operate at low enough combustion temperatures to permit compliance with the NO_x specification. In addition to these two primary combustion processes, a conventional furnace with stack gas cleanup was also evaluated for the advanced steam cycle, and a direct coal furnace with stack gas cleanup was evaluated for the closed-cycle inert gas MHD cycle.

The furnace concept utilized as a base case furnace for most closed cycles was the atmospheric fluidized bed. This furnace is shown schematically in Figure 1-1. The energy to the prime cycle is extracted from the fluidized bed through heat exchange tubes in the bed and in the convective space above the bed. The bed material consists of combusting coal, ash, and limestone. The latter is utilized for sulfur capture. The combustion process was assumed controlled to 1550 F (1117 K) in order to maximize the sulfur removal. If the temperature is greater than 730 F (611 K), the exhaust gases exiting the furnace go through a high-temperature air preheater. If the temperature is equal to 730 F (611 K) it goes directly to an electrostatic precipitator for fly ash removal. The exhaust gas exits the electrostatic precipitator and enters a low-temperature air preheater. The low-temperature air preheater drops the exhaust gas temperature to less than 300 F (422 K) before exiting in the stack. Both a forced and induced draft fan were employed for air supply.

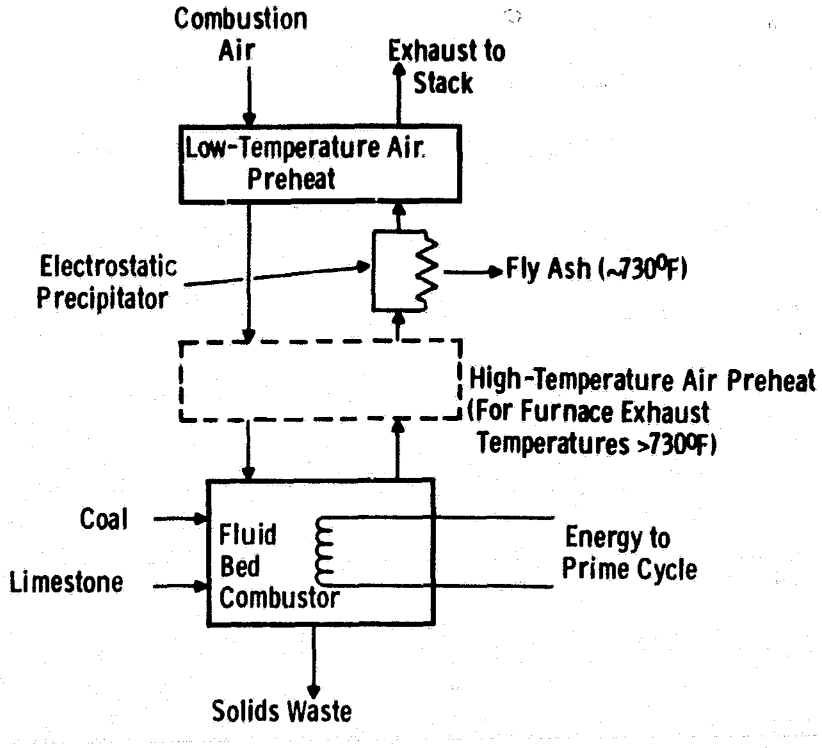

Figure 1-1. Furnace Configuration for Direct Coal Combustion Atmospheric Fluidized Bed

The second direct coal combustion scheme is shown in Figure 1-2. In this scheme, the fluidized bed operates at 10 atmospheres. The pressurization system is an industrial gas turbine. The energy to the prime cycle is again taken from the fluidized bed. However, in order to operate with maximum gas temperature entering the gas turbine, the heat exchange tubes for the prime cycle are located only within the fluidized bed. The combustion process was assumed controlled to 1650 F (1172 K) bed temperature with a 1600 F (1144 K) turbine inlet temperature. The system was operated at 20 percent excess air. Two techniques were employed for reducing the turbine exhaust temperature; both are shown in Figure 1-2. The first employs a regenerative heat exchanger which preheats the combustion air and the second employs an economizer which was utilized in the advanced steam case to perform feedwater heating duty. The pressurizing gas turbine generated more than enough energy to drive the furnace air supply compressor. The excess energy was removed in an electrical generator. The exhaust gas leaving the fluidized bed must be cleaned up at temperature (~1600 F [1144 K]) to a quality which is acceptable to the gas turbine. The scheme which was employed consists of cyclones and granular bed filters. However, there is no equipment experience to indicate that the cleanup scheme can produce gas turbine quality gas. This high-temperature gas cleanup is a major development problem for pressurized fluidized beds.

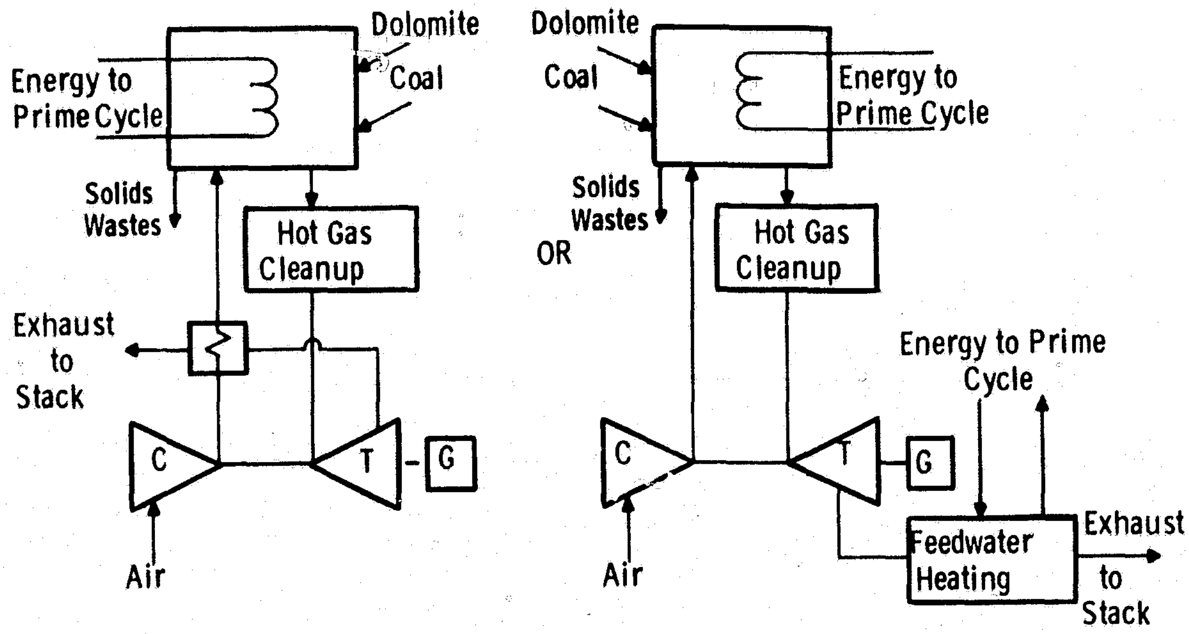

Figure 1-2. Furnace Configuration for Pressurized Fluidized Bed (Direct Coal Combustion)

Clean gases used to supply energy for a closed cycle were combusted in a pressurized furnace. In this scheme, a pressurizing gas turbine was again employed as an air supply, with the exhaust gas from the furnace being expanded in a gas turbine to supply the compressor power demand and to generate extra electricity. The pressurized furnace was operated 8 atmospheres. The clean gas ensured compliance with the SO_X specification. A staged combustion process was employed to control the combustion temperature and comply with NO_X specifications. The energy was transferred to the prime cycle by forced convection to heat exchanger tubes. Twenty percent excess air was employed in this combustion process. Two clean gases were utilized, high-Btu gas delivered "over-the-fence" to the power plant and low-Btu gas produced from coal, on site, in a low-Btu gasifier integrated with the furnace system. The high-Btu gas configuration is shown in Figure 1-3. In this system, the turbine exhaust is cooled by heating the combustion air in a recuperative heat exchanger. The low-Btu gas case is shown in Figure 1-4. In this case, the exhaust from the gas turbine is used to generate steam in a heat recovery steam generator. This steam generator features two steam drums operating at different pressures. One drum supplied saturated steam to the gasifier, the other superheated steam to a bottoming steam turbine. Additional gasifier steam demand is supplied from a steam turbine extraction point. The gasifier is a fixed bed type. The low-Btu gas is produced by a reaction of coal, steam, and air. The air supply to the gasifier is obtained from the gas turbine compressor. A booster air compressor is required, and this is driven by a steam turbine operating on extraction steam from the bottom-

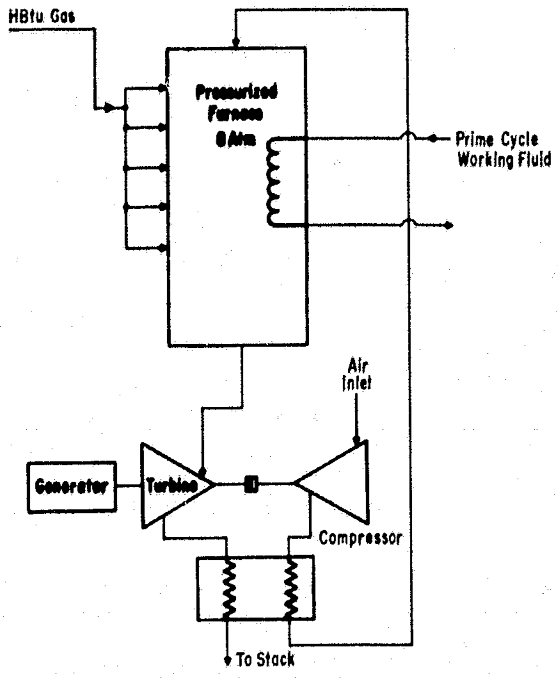

Figure 1-3. Furnace Configuration for Pressurized Furnace-High Btu (Clean Gas Combustion)

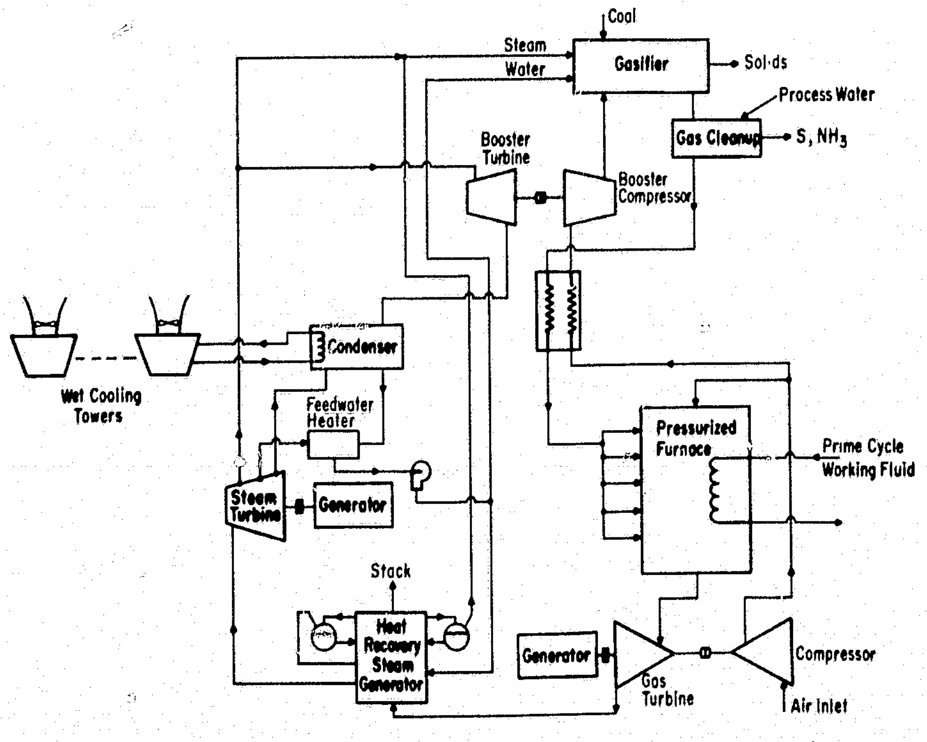

Figure 1-4. Furnace Configuration for Pressurized Furnace with Integrated Low-Btu Gasifier (Clean Fuel Combustion)

ing steam turbine. A low-temperature gas cleanup scheme was employed to remove the S and NH_3 from the low-Btu gas before combustion in the pressurized furnace. Although this integration scheme is complex, it was used to achieve an efficient plant. The components and integration scheme are similar to another energy conversion system employing low-Btu gasification, the gas turbine combined cycle.

A wide variety of fuels and combustor types were employed as energy sources for the various prime cycles. These combinations are shown in Table 1-4 for both the open and closed cycles.

Table 1-4

COMBUSTION AND FUEL COMBINATIONS

Cycles	Fuels	Combustion Type
Open cycles		
Gas turbines		
Simple/regenerative	Clean fuels, solvent refined coal (SRC)	Direct
Combined	Clean fuels, SRC, Integrated Low-Btu	Direct
MHD	Coal, SRC	Direct
Closed cycles		
Gas turbine	Coal, Low-Btu, High-Btu	Atmospheric fluidized bed Pressurized furnace Pressurized fluidized bed (recuperative)
Supercritical CO_2	Coal, Low-Btu, High-Btu	Atmospheric fluidized bed Pressurized furnace Pressurized fluidized bed (recuperative)
Advanced steam	Coal, Low-Btu, High-Btu SRC	Atmospheric fluidized bed Pressurized furnace Pressurized fluidized bed Pressurized fluidized bed (recuperative) Conventional furnace
Liquid metal topping	Coal, Low-Btu, High-Btu	Atmospheric fluidized bed Pressurized furnace Pressurized fluidized bed (recuperative)
Inert gas MHD	Coal, SRC	Direct fired
Liquid metal MHD	Coal, Low-Btu, High-Btu	Atmospheric fluidized bed Pressurized furnace Pressurized fluidized bed (recuperative)
Fuel Cells		
Low-Temperature	High-Btu, H_2	—
High-Temperature	Low-Btu (free-standing)	—

The major environmental intrusions which were considered were thermal and exhaust gas emissions. The thermal pollution of water bodies was minimized by employment of wet or dry cooling towers. The exhaust emissions were to be controlled to a standard which was specified by NASA. This standard is shown in Table 1-5. The emission control technique for each cycle is indicated in Table 1-6. The equipment design and cycle operating conditions were set to permit the emission standards to be met.

Table 1-5

EMISSION CHARACTERISTICS STANDARDS

Pollutant	Fuel	Standard ($lb/10^6$ Btu Input)
SO_X	Solid	1.2
	Liquid	0.8
	Gaseous	0.2
NO_X	Solid	0.7
	Liquid	0.3
	Gaseous	0.2
Particulates	All Fuels	0.1

Table 1-6

ENVIRONMENTAL INTRUSION CONTROL TECHNIQUES

Cycles	Control Techniques
Open cycles	
Gas turbine	Water injection, clean fuels
MHD	Seed material, residence time
Closed cycles	
Gas turbine	During combustion, clean fuels
Supercritical CO_2	During combustion, clean fuels
Advanced steam	During combustion, clean fuels stack gas cleanup
Liquid metal topping	During combustion, clean fuels
Inert gas MHD	Clean fuels, stack gas cleanup
Liquid metal MHD	During combustion, clean fuels
Fuel cells	Clean fuels

The open cycles feature direct combustion with combustion gases, the cycle working fluid being exhausted. In open-cycle gas turbines, the SO_X criterion was met by employment of clean fuels. The fuel bound nitrogen was removed or assumed to be removed in the fuel processing system. In the case of the semi-clean liquid (SRC), the as-delivered fuel had too high a nitrogen content to enable adherence to the emission specifications when this fuel was employed in an open-cycle gas turbine. The thermal NOX was controlled by water or steam injection into the combustion chamber or in the case of low Btu gas, by water vapor entering with the fuel. In open-cycle MHD, the SO_X criterion was met by combining the sulfur with the MHD seed material. The NO_X criterion was met by providing sufficient residence time at a temperature that established equilibrium at an acceptable NO_X level.

In the closed-cycle furnaces, the SO_X criterion was met by employing either clean fuels, in-bed combustion cleanup (e.g., fluidized beds) or stack gas cleanup. The NO_X criterion was met by combustion control, such as staged combustion.

In fuel cells, clean fuels were employed as energy sources. This permitted compliance with the emission standards.

STANDARD POWER PLANT COMPARISON

The majority of the utility's baseload from fossil fuels is presently carried by steam power plants with conventional furnaces. In order to establish a calibration basis for the review of the advanced energy conversion system results which are presented in the following sections, a conventional furnace, steam power plant was analyzed. The assumption and calculation techniques which were used in this analysis are the same as were utilized in all advanced cycle evaluations.

The steam cycle was assumed to have steam throttle conditions of 3500 psi (2.41×10^7 N/m^2) 1000 F (811 K) with a single reheat to 1000 F (811 K). The results of the evaluation on this steam cycle are shown in Table 1-7. The first case employs an atmospheric fluidized bed, with in-bed combustion sulfur removal, and a wet mechanical draft cooling tower. (This was a parametric variation of the advanced steam cycle.) The efficiency of this system was estimated to be 36.5 percent with a 29.8 mills/kWh cost of electricity. In the second case, a convention radiant furnace with stack gas cleanup for the sulfur capture is evaluated. The stack gas cleanup system which was employed in Task I of this study (discussed in Volume III) did not require steam extraction for stack gas reheat. The efficiency of the plant therefore went up slightly to 37.1 percent due to a reduced furnace fan power requirement. The cost of electricity however increased to 31.9 mills/kWh due to higher capital costs. In the third case, the cooling towers were replaced with once through cooling and the scrubber is removed from the stack gas cleanup system. The efficiency in this case goes up to 37.6 percent and due to a significant reduction in capital cost the cost of electricity is reduced to 28 mills/kWh.

Table 1-7

STEAM POWER PLANT COMPARISON

<u>As Analyzed by Study Techniques</u>

- Cleanup during combustion—Atmospheric fluidized bed

  Steam cycle: 3500 Psi, 1000/1000 F

  Wet mechanical draft cooling towers

Overall efficiency (%)	36.5
Capital cost ($/kWe)	610
Cost of electricity (mills/kWh)	29.8

- Conventional furnace

  Steam cycle: 3500 psi, 1000/1000 F

  Stack gas cleanup
  Wet mechanical draft cooling towers

Overall efficiency (%)	37.1
Capital cost ($/kWe)	690
Cost of electricity (mills/kWh)	31.9

- Conventional furnace

  Steam cycle: 3500 psi, 1000/1000 F

  Limited stack gas cleanup
  Once through cooling

Overall efficiency (%)	37.6
Capital cost ($/kWe)	570
Cost of electricity (mills/kWh)	28.0

<u>As Built</u>

- Bull Run Plant (TVA)—Conventional furnace

  Steam cycle: 3500 psi 1000/1000 F

  Limited stack gas cleanup
  Once through cooling
  (Values averaged over 1971 operation)

Efficiency	37.7

The efficiency of these power plant concepts was compared with data from an "As Built" plant. This plant was the Bull Run Plant on the TVA system, which operates with conditions similar to the third condition. The 1971 FPC data list an average overall efficiency for this plant of 37.7 percent. Although this average reported efficiency does contain brief periods for startup and part load operation where the efficiency would be less than normal, it does provide a conformation check on the results for "As Analyzed" efficiency.

With the study groundrules as defined and the analysis procedures applied, a steam power plant with conventional conditions (3500 psi/1000 F/1000 F [2.41×10^7 N/m^2 811 K/811 K]) designed to meet the environmental standards would cost between \$600/kW and \$700/kW. This plant would operate at an overall efficiency of between 36 and 37 percent and produce electricity at a cost of about 30 mills/kWh. These numbers should be used as a guideline as the results for the advanced energy conversion systems are reviewed.

REFERENCE

1. Power Plant Capital Costs: Current Trends and Sensitivity to Economic Parameters, Report WASH-1345, U.S. Atomic Energy Commission Contract W-7405-eng-26, Oak Ridge National Laboratory; Contract AT (11-1)-2226, United Engineers and Constructors Inc., U.S. Government Printing Office, October 1974, 74 pp.

Section 2

CYCLE ANALYSIS, RESULTS, AND DISCUSSION

2.1 OPEN-CYCLE GAS TURBINE

DESCRIPTION OF CYCLE

The schematic of the simple cycle gas turbine is presented in Figure 2.1-1, and the schematic for the recuperative cycle gas turbine is presented in Figure 2.1-2. Figure 2.1-3 shows the addition of an organic fluid bottoming cycle to the recuperative gas turbine. The air-cooled gas turbine components all have airflows of 570 lb/s, (2.05 million lb/hr [259 kg/s]), resulting in nominal outputs of 85 MW at a turbine inlet temperature of 2200F (1480 K). The organic bottoming cycle adds 24 MW to the gross generation.

Component Descriptions

Compressor. Ambient air is pressurized in a single shaft, multistage, axial flow compressor. Compressor inlet air is filtered and passed through silencers prior to entering the compressor inlet plenum.

Combustor. Gaseous or liquid fuels are burned with pressurized compressor discharge air in the gaseous recirculating core (primary zone) of the combustor. Secondary air flowing over the outside of the combustion liner is introduced through holes and louvers to provide dilution and liner cooling.

The design of the combustor must be different to burn low-, intermediate-, or high-Btu gas or to burn specific liquid fuels. The combustion liners remain coolest burning gases and become successively hotter burning distillate and then residual oil as a result of the greater emissivity of their flames.

Steam or water is injected into the combustor primary zone to reduce NO_x production. Steam or water injection is not required for low- or intermediate-Btu gases because of their low stoichiometric flame temperature.

Turbine. A single shaft, three-stage turbine is used to drive the air compressor and electric generator. A four-stage turbine is used for compressor pressure ratios of 20.

The hot pressurized combustion gas is expanded and accelerated through a nozzle before impinging on the rotating buckets. The first-stage nozzle (typically a Ni base superalloy, such as IN-738) sees the highest temperature. A portion of the air from the compressor discharge is used to reduce nozzle metal temperatures to a level which will provide adequate strength and oxidation resistance. This cooling air mixes with the combustion gas at the nozzle exit and is expanded through the turbine. Turbine

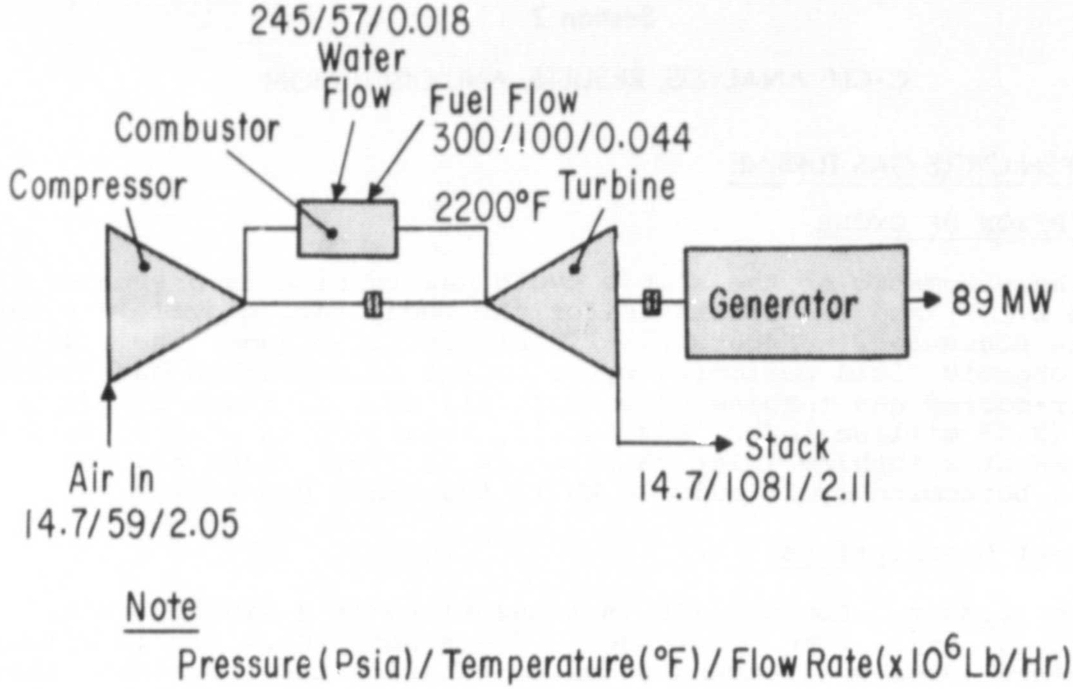

Figure 2.1-1. Open-Cycle Gas Turbine—Simple Cycle

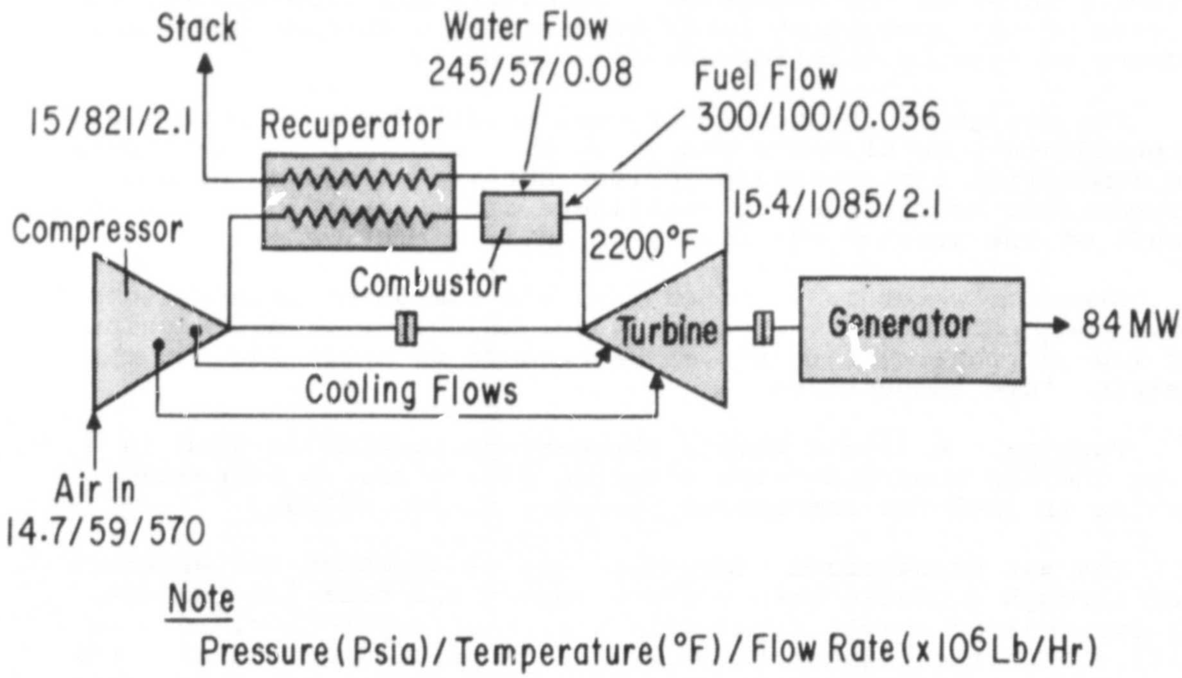

Figure 2.1-2. Open-Cycle Gas Turbine—Recuperative Cycle

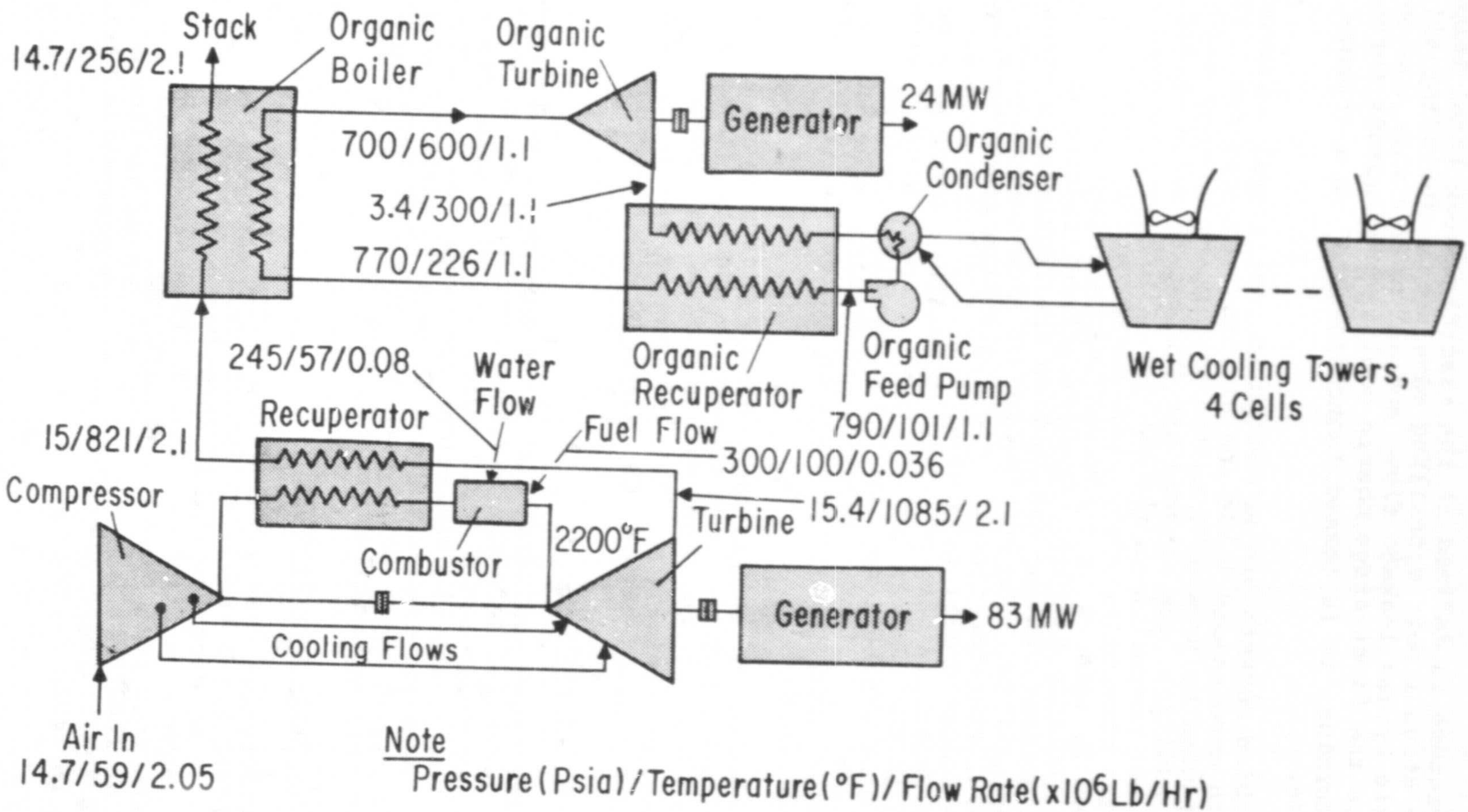

Figure 2.1-3. Open-Cycle Gas Turbine—Recuperative with Organic Bottoming

firing temperature is defined as the mass-flow-mean total temperature at the stage 1 nozzle trailing edge plane, including all stage 1 nozzle bypass leakage flows and all nozzle vane cooling flows. Since the first-stage nozzle cooling air expands through the entire turbine, it is termed "nonchargeable" in a thermodynamic sense.

The turbine buckets are subjected to centrifugal stresses, combined with aerodynamic loads and elevated temperatures. Vibrations are induced by proximity to the stationary nozzles and struts. In addition, the buckets are subject to solid particle erosion and hot corrosion in the gas jet. Carryover of fuel ash, fuel treatment chemicals, sand, etc., causes erosion and fouling. Hot corrosion is induced by carryover and condensation of alkali metal salts (notably Na_2SO_4). Specifications limiting the carryover of contaminants to the gas turbine are required to maintain adequate design life. The combination of high temperatures, centrifugal load, vibratory forces, and aggressive atmosphere makes the turbine bucket a critical design area for gas turbines. Buckets are generally cast of a high strength, nickel base superalloy, such as IN-738. At turbine inlet temperatures in excess of 1800 F (1260 K) extractions from the compressor are used for bucket and nozzle cooling. Exhaust temperature is defined as the mass-flow-mean total temperature of the exhaust flow, including all dilution flows.

In the water-cooled gas turbine high purity water is used to cool the buckets, nozzles, and transition pieces. Bucket cooling passages are very narrow; high purity coolant is required to avoid plugging and/or erosion. Gas turbine exhaust is ducted through silencers and a stack before its return to the atmosphere.

Simple Cycle Gas Turbine

The five cases evaluated for the simple cycle gas turbine of Figure 2.1-1 all are 3600 RPM machines burning high-Btu gas delivered at 600 psia, 100 F (4.14 MN/m^2, 311 K). Base case cycle conditions and parametric variations are presented in Table 2.1-1. Except for the water-cooled turbine of Case 5, with 700 lb/s (318 kg/s) airflow, all other units have 570 lb/s (259 kg/s) airflow, and utilize only air cooling for the turbine.

Recuperative Cycle Gas Turbine

The thirty-two cases evaluated for the recuperative gas turbine of Figure 2.1-2 and 2.1-3 are all derived from the air-cooled 3600 RPM unit of Case 6 with 570 lb/s (259 kg/s) airflow and an output of 83 MW when firing at 2200 F (1480 K). A single case at 25 MW is examined for operation at 5100 RPM. A single case is evaluated at 1800 RPM to produce 328 MW with an airflow of 2280 lb/s (1030 kg/s). Base case conditions and parametric variations are shown in Table 2.1-2.

Table 2.1-1

SIMPLE CYCLE GAS TURBINE EVALUATION

Parameter	Base Case	Variations
Power output (MW)	87	111 to 212
Number of coals	1	-
Coal conversion process	High-Btu gas	-
Turbine inlet temperature (°F)	2200	2600, 3000
Compressor pressure ratio	12	16
Airflow (lb/s)	570	700 for water-cooled

Table 2.1-2

RECUPERATIVE CYCLE GAS TURBINE EVALUATION

Parameter	Base Case	Variations
Power output (MW)	83	25,331, 60 to 111
Number of coals	1	2
Coal conversion process	High -Btu gas	Liquid COED, SRC
Turbine inlet temperature (°F)	2200	1800, 2000, 2600
Compressor pressure ratio	12	8, 16
Compressor airflow (lb/s)	570	172, 2280
Recuperator effectiveness	0.85	0.8, 0.9, 0.95
Recuperator pressure drop, $\Delta P/P$	0.05	0.03, 0.07, 0.09

The recuperator preheats the air before combustion, with a net reduction of fuel required to reach a specified firing temperature. This benefit diminishes with high pressure ratios and hotter compressor discharge air temperature. High-Btu gas was the designated fuel for most cases; liquid COED and liquid solvent refined coal (SRC) were considered for specific cases.

Rationale for Point Variations

The simple cycle gas turbine was of primary interest for peaking service where fuel cost has low economic impact. High-Btu gaseous fuel was selected for all cases except for consideration of liquid COED fuel and liquid SRC in two instances. The power level results from the assigned airflow, which represents a reasonable level for 3600 RPM gas turbine units. The water-cooled gas turbine can operate at greater stress levels than air-cooled units because of the greatly reduced temperatures of the rotors and turbine buckets. By bringing the airflow to 700 lb/s for the

water-cooled turbine, the design margins are made comparable to those experienced on an air-cooled turbine with 570 lb/s airflow. A lesser rating was examined at 25 MW, and a single 1800 RPM unit was evaluated that would replace four of the standardized units. The base case at 2200 F (1480 K) and 12 pressure ratio is a reasonable extension of current practice. The 2600 F (1700 K) cases are an extension of air cooling while 3000 F (1920 K) would represent an extreme for air cooling but a moderate level for water cooling. Reduced temperatures of 1800 F (1260 K) and 2000 F (1370 K) were considered for the recuperative cycle. These reduced temperatures provide a comparison with current state-of-the-art recuperative gas turbines. Ceramic stationary parts, combustor and transition piece and nozzles, were evaluated for 2600 F (1700 K) at a 16 pressure ratio. It was expected that ceramics would be applicable to stationary parts well in advance of their successful application to highly stressed rotating parts like turbine buckets. Their best economy and hence their largest payoff will be found at the highest firing temperatures, where the air cooling proves to have the greatest parasitic penalty. Table 2.1-3 indicates the grouping of temperature and pressure variations about the base case for recuperative cycles.

Recuperator effectivenesses of 0.8, 0.9, and 0.95 were explored about the base value of 0.85. The recuperator sum pressure loss ratios were 0.03, 0.07, and 0.09, and the base case was 0.05. Inlet air temperatures of 20 F (266 K) and 93 F (308 K) were evaluated as well as the base value of 59 F (288 K) to determine the sensitivity to ambient temperature.

Table 2.1-3

VARIATIONS IN TEMPERATURE AND PRESSURE RATIO FOR RECUPERATIVE CYCLE GAS TURBINES

Pressure Ratio	Maximum Temperature, ^{o}F			
	1800	2000	2200	2600
8	-	-	(20)*	(17)
12	(14)	(15)	(6)	(16)
16	-	-	(21)	(18) (19,CN)**

* Numerals in ( ) are case numbers.

** CN = ceramic nozzles and parts

For the recuperative gas turbine with a Fluorinol-85 bottoming cycle, the lowest acceptable temperature for the gas turbine exhaust was 250 F (394 K). The pinch-point temperature difference between heating organic fluid and cooling exhaust gas has a dominant influence on heat exchanger size and cost. Values of 30, 50, and 70 F (16.7, 27.8, and 38.9 K difference) were examined with base conditions and at a value of 30 F (27.8 K difference)

for 1800 F (1260 K) firing temperatures. Organic fluids generally have poorer heat transfer properties than water. The typical pinch point for water to gas in heat recovery steam generators is 30 F to 50 F (16.7 to 27.8 K difference). An optimum value would be expected in the spread values selected for consideration. Organic cycle condensing was considered to be at 99.1 (310 K) with dry cooling towers except for one case at 92 F (306 K) for a wet cooling tower.

ANALYTICAL PROCEDURES AND ASSUMPTIONS

It has often been convenient to assign specific fixed performance parameters for gas turbine components and then not vary them in a given study. This is an oversimplification. A more elaborate process involves conforming the performance to proven test results for both specific detailed performance—like the cooling capability of an airstream cooling a bucket—and for overall performance—like compressor polytropic performance. Such conformance results in a more accurate prediction of expected performance. However, these methods tend to reveal explicit proprietary design details that cannot be made public in this report. Nevertheless this method was followed for this study. The methodology will be described in detail, but the numerical values used will not be detailed so as to preserve the confidential nature of the data base.

Open-Cycle Component Assumptions

Inlets. The simplest type of inlet for a heavy-duty gas turbine consists of duct work and a trash screen and imposes a pressure drop of about 2 in. H_2O (498 N/m^2). Further inlet air treatment can consist of weather louvers, up to 3 stages of filtering, and various amounts of silencing, all together imposing a worst case pressure drop of about 5 in. H_2O (1240 N/m^2).

In this study, an inlet pressure drop of 4 in. H_2O (995 N/m^2) has been assumed, which is representative of a well-silenced inlet with two stages of filtering.

Compressors. Field measurements were correlated for gas turbine compressors with inlet airflow in excess of 200 lb/s (90.7 kg/s). These data form the basis for a schedule of polytropic efficiency and adiabatic efficiency as a function of pressure ratio. At reduced airflow, under 200 lb/s (90.7 kg/s), corrections are applied to account for increased end wall effects. The rotational speed of the gas turbine set related to airflow is indicated in Table 2.1-4.

Combustors. A single combustion efficiency of 98 percent based on lower heating value and a single combustor pressure drop of 3 percent were assigned. These assumptions imply different design geometry for the different fuels and temperature rises required.

Table 2.1-4

GAS TURBINE COMPRESSOR SPECIFICATIONS

Turbine Descriptor	RPM	Airflow (lb/s)
Air-cooled, normal	3600	570
Air-cooled, 25 MW	5100	172
Air-cooled, 331 MW	1800	2280
Water-cooled	3600	700

Turbines. In place of assigning an overall turbine adiabatic efficiency, the work output and the temperature effects were evaluated stage by stage. The single stage air-cooled turbine was assigned a constant total-to-total adiabatic efficiency. The inlet total temperature was evaluated for the combination of mainstream flow and cooling airflows entering before the stage nozzle. The air coolant flow to a particular stage was treated as if it merged with the mainstream just after that stage and produced no power. The schedule of cooling airflow to each stage depended on the total temperature at that stage. The last-stage turbine diffuser loss was assigned a constant value equal to current design experience.

The form of the schedules for chargeable cooling flow, the air admitted for cooling after the first-stage nozzles, is shown in Figure 2.1-4. No coolant flow would be needed below 1350 F (1010 K) turbine inlet temperature. The current industrial gas turbine internal passage air cooling would prove effective to limits in excess of current firing temperatures. The film and internal air cooling as practiced on aircraft gas turbines would prove more effective at higher firing temperatures. Advanced film cooling, as currently under development for advanced aircraft gas turbines, should permit even higher firing temperatures in the future.

Rather than designate the specific mode of cooling to be used in each instance, a single schedule was synthesized and used in this study to express the chargeable compressor extraction airflow to each turbine stage.

First-stage nozzle cooling air and all bucket cooling air are extracted at compressor discharge. Other nozzle cooling air and wheel space cooling and blockage air are extracted at a compressor stage whose pressure is 20 percent above the pressure at which the cooling air re-enters the main gas path. For the water-cooled turbine air is still required for wheel space sealing or blockage.

The heat loss due to water cooling was calculated for each element in the cooled gas path (i.e., nozzle blading and wall interstage passage, bucket, bucket shroud, wall, and platform, transition piece, and turbine exit diffuser). This heat loss was

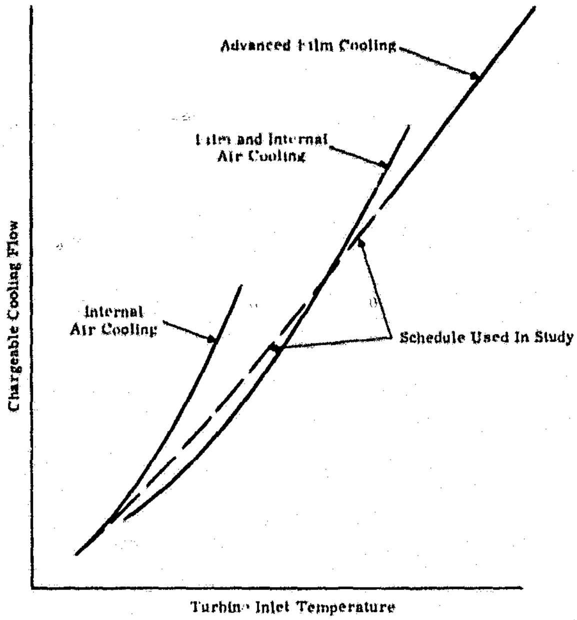

Figure 2.1-4. Form of Cooling Air Schedule

based on gas temperature, metal surface temperature, and stage geometry. Changes in stage efficiency due to nonadiabatic bucket flow and increased tip leakage due to the low temperature of the leakage at the wall (thermal boundary layer effect) were also considered.

The work required to bring the bucket water coolant up to bucket tip speed and the energy recovered when this coolant was expanded through the bucket tip nozzles were calculated and considered as part of the turbine output. Blockage air and unrecovered bucket coolant were mixed between stages to determine the gas properties and temperature entering the subsequent turbine stage.

When ceramic nozzles and stationary hot gas path parts were designated, the nozzles, the transition piece, and the combustor were deemed to be uncooled.

The turbines comprised three-stage units for compressor pressure ratios of 8, 10, 12, and 16. Four stages were the design basis for a pressure ratio of 20.

Recuperators. The recuperators used plate-fin surfaces in a shell and tube configuration. The performance of the recuperator was determined by the assigned effectiveness and sum of pressure losses. The design basis used standard relationships for friction factor and heat transfer coefficient as related to the local Reynolds number of the flow. The base effectiveness was 85 percent with a 5 percent sum of pressure losses.

Organic Bottoming Cycle. The applicable criteria for the organic bottoming cycle and the cycle performance with the

recuperative gas turbine are presented in Section 4, "Bottoming Cycles." The pressure loss added to the gas turbine exhaust path was assigned as 10 inches of water ($2.5 \times 10^3 N/m^2$) with a single case at 20 inches ($5 \times 10^3 N/m^2$) to explore sensitivity to this factor. Fluorinol-85 was the organic fluid used for recuperative cycle gas turbine bottoming.

Balance of Plant. The skid mounted gas turbine units are field connected with a minimum of site labor to form a self-sufficient power unit capable of remote starting and stopping and unattended operation. Only when the organic bottoming cycle is added does this item become significant.

Exhaust Systems. The exhaust pressure loss incurred from the exhaust plenum flange static pressure to ambient, including silencing, was 5 in. H_2O (1240 N/m^2).

Other Components. Generator overall efficiency is assumed to be one level for units in the 60 MW class and a lower level for units in the 25 MW class.

Accessory losses for bearings and accessory drives as a percentage of generator output were assumed to vary linearly with unit capacity.

Gas Properties. The computer program used for thermodynamic cycle calculations synthesized local mixture properties from the actual gas composition based on the National Bureau of Standards Circular 564 and the ASME Steam Tables of 1967. The subdivision of gas composition was treated as water or water vapor, carbon dioxide, air, and oxygen depletion. Trace elements like sulfur or nitrogen compounds were evaluated separately for emissions evaluation.

DESIGN AND COST BASIS

The gas turbine units would be delivered assembled on foundation skids that minimize the site labor and facilitate the connection to necessary facilities. For the liquid fuels such as COED or SRC a modification of the residual oil preconditioning system would be furnished mounted on its own foundation skid. This fuel cleaning and treating unit was sized and costed on a basis equal to experience with residual oil units. The basic gas turbine cost includes inlet and exhaust systems; fuel regulation and control; compressor, combustor, and turbine; generator, exciter, and electrical control; fuel conditioning where applicable; and the gas turbine recuperator cost. The organic fluid heat exchangers described in Section 4 were designed on the basis of serrated finned surfaces to exchange exhaust gas heat into the organic fluid. The organic recuperator that desuperheats vapor to be condensed by heating the feed liquid used plate-fin surface design. The condenser for organic vapors used tubes finned on the organic vapor side. The sizing of the required surfaces followed conventional practice as outlined in Section 4.

The costs for these organic fluid heat exchangers were determined using $7/linear ft for the 1-in. diameter tubes of the organic boiler and the organic fluid exhaust desuperheater-recuperator. The organic fluid condenser cost was evaluated at $8/ft^2 of condenser tube base area.

The specific designs and sizes of the organic fluid pumps and turbines were evaluated as outlined in Section 4. The pump prices were derived from vendor quotations for pumps of similar physical size. The turbine costs were derived by extrapolating known costs for several smaller organic fluid turbines.

The balance-of-plant costs were derived from known installation costs for gas turbines and their recuperators. Where organic turbines were applied, their field erection costs were estimated from knowledge of costs for comparable physical sizes of steam heat exchangers, condensers, and small auxiliary generator drives.

The actual final cost of the base case recuperative gas turbine was determined by a detailed analysis. Thereafter the costs of parametric cases were determined by evaluating their cost differentiation from the base case.

RESULTS

The simple cycle gas turbine is summarized in Table 2.1-5 for a compressor pressure ratio of 12 firing to 2200 F (1480 K) first-turbine stage total temperature with high-Btu gas. The low installed cost of $73/kW for the gas turbine and $125/kW for the total plant produced a cost of electricity of 36 mills/kWh; this is just ten percent greater than the base cycle cost for advanced steam. The modest cycle efficiency of 29.6 percent was degraded to 14.9 percent by the inefficiency in producing high-Btu gas from coal. Water was required for suppression of NO_X production during combustion.

The base case recuperative cycle gas turbine is summarized in Table 2.1-6 for conditions identical to the simple cycle gas turbine. Increased pressure losses reduced the net output from 87 MW to 83 MW while the efficiency improved from 29.9 percent to 34.4 percent. Notwithstanding the high cost of the fuel gas, the 33.2 mills/kWh cost of electricity equaled that of the base case for advanced steam. The basic turbine cost would be $100/kW and the plant $166/kW mainly as a result of the recuperator cost.

Table 2.1-7 presents the full array of parametric points and results. Cases 1 through 5 are simple cycle machines, as illustrated in Figure 2.1-1. Case 6 is the base case recuperative gas turbine. Cases 6 through 29 are recuperative gas turbines, as illustrated in Figure 2.1-2. Cases 30 through 37 (32 and 33 were deliberately deleted) are recuperative gas turbines with organic bottoming cycles, as illustrated in Figure 2.1-3. All but Case 36 use dry cooling towers.

Table 2.1-8 presents the capital cost distributions for these cases.

Table 2.1-5

SUMMARY SHEET
OPEN-CYCLE GAS TURBINE BASE CASE (SIMPLE CYCLE)

CYCLE PARAMETER

Net Power Output (MWe)	87
Coal Type and Conversion Process	Illinois No. 6 HBtu
Prime Cycle	
Turbine inlet temperature (°F)	2200
Compressor pressure ratio	12
Air inlet temperature (°F)	59

PERFORMANCE AND COST

Thermodynamic efficiency (percent)	29.9
Powerplant efficiency (percent)	29.6
Overall energy efficiency (percent)	14.9
Plant capital cost ($ x 10^6)	11
Plant capital cost ($/kWe)	125
Cost of electricity (mills/kWh)	36.0

NATURAL RESOURCES

Coal (lb/kWh)	2.12
Water (gal/kWh)	
Total	0.025
Cooling	0
Processing	0
Makeup	0
NO_x suppression	0.025
Stack gas cleanup	0
Land (acres/100 MWe)	2.87

ENVIRONMENTAL INTRUSION

	Lb/10^6-Btu Input (fuel)	Lb/kWh Output
SO_2	0.009	0.104×10^{-3}
NO_x	0.20	2.319×10^{-3}
HC	--	--
CO	--	--
Particulates	--	--

	Btu/kWh
Heat to water	0
Heat, total rejected	6925
Wastes	--

MAJOR COMPONENT CHARACTERISTICS

	Unit or Module					
Major Component	Size (ft) (W x L (or D) x H)	Weight (lb) (x 10^6)	Cost ($ x 10^6)	Units Required	Total Cost ($ x 10^6)	$/kW Output
Gas turbine-compressor-combustor-generator	11 x 62 x 14	0.56	6.3	1	6.3	72.9

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PARAMET

Parameters	Simple Cycle										
	Case 1*	2	3	4	5**	6*	7	8	9	10	11
Power Output (MWe)	87	111	115	130	212	83	83	83	80	79	25
Coal and Conversion Process	Ill. #6 HBtu →	→	→	→	→	→	N. D. HBtu	Mont HBtu	Ill. #6 Liq coed	Ill. #6 Liq SRC	Ill. #6 HBtu →
Prime Cycle											
Turbine inlet temperature (^{o}F)	2200	2600 →	†† →	3000 →	→	2200 →	→	→	→	→	→
Compressor pressure ratio	12	16 →	→	→	→	12 →	→	→	→	→	→
Recuperator efficiency	--	--	--	--	--	0.85 →	→	→	→	→	→
Recuperator pressure drop ($\Delta p/p$)	--	--	--	--	--	0.05 →	→	→	→	→	→
Air inlet temperature (^{o}F)	59 →	→	→	→	→	→	→	→	→	→	→
Added gas side (Δp in. H_2O)	--	--	--	--	--	--	--	--	--	--	--
Bottoming Cycle											
Superheat to inlet gas temperature differential (^{o}F)	--	--	--	--	--	--	--	--	--	--	--
Boiler pinch point temperature differential (^{o}F)	--	--	--	--	--	--	--	--	--	--	--
Heat Rejection											
T condense (^{o}F)	--	--	--	--	--	--	--	--	--	--	--
Actual Powerplant Output (MWe)	87	111	115	130	212	83	83	83	80	79	25
Thermodynamic Efficiency (percent)	29.9	32.2	32.4	31.9	32.6	34.8	34.8	34.8	36.1	36.6	34.1
Powerplant Efficiency (percent)	29.6	31.9	32.1	31.7	32.5	34.4	34.4	34.4	35.7	36.1	33.8
Overall Energy Efficiency (percent)	14.9	16.1	16.2	16.0	16.4	17.3	17.3	17.3	20.0	28.2	17.0
Coal Consumption (lb/kWh)	2.12	1.97	1.95	1.98	1.93	1.82	2.86	2.20	1.58	1.12	1.86
Plant Capital Cost ($ million)	11	15	15	18	21	14	14	14	15	15	5
Plant Capital Cost ($/kWe)	125	138	134	134	99	166	166	166	189	190	215
Cost of Electricity, Capacity Factor = 0.65											
Capital (mills/kWh)	4.0	4.4	4.2	4.2	3.1	5.3	5.3	5.3	6.0	6.0	6.8
Fuel (mills/kWh)	30.0	27.8	27.6	28.0	27.3	25.8	25.8	25.8	24.9	17.0	26.3
Maintenance and operating (mills/kWh)	2.0	1.6	1.5	2.0	1.7	2.1	2.1	2.1	2.6	2.7	2.9
Total (mills/kWh)	36.0	33.8	33.4	34.3	32.1	33.2	33.2	33.2	33.5	25.7	35.9
Sensitivity											
Capacity factor = 0.50 (total mills/kWh)	37.8	35.6	35.1	36.2	33.6	35.4	35.4	35.4	36.1	28.3	38.8
Capacity factor = 0.80 (total mills/kWh)	34.8	32.7	32.3	33.1	31.2	31.8	31.8	31.8	31.9	24.1	34.1
CapitalΔ = 20 percent (Δmills/kWh)	0.8	0.9	0.8	0.8	0.6	1.1	1.1	1.1	1.2	1.2	1.4
FuelΔ = 20 percent (Δmills/kWh)	6.0	5.6	5.5	5.6	5.5	5.2	5.2	5.2	5.0	3.4	5.3
Estimated Time for Construction (years)	1	1	1	1	1	1	1	1	1	1	1
Estimated Date of 1st Commercial Service (year)	1980	1983	1983	1986	1986	1980	1980	1980	1980	1980	1980

*Base Case
**Water-cooled turbine
***Four units
†1800 rpm
††Ceramic nozzle

DCT = Dry cooling tower
HBtu = High Btu
Ill = Illinois
Liq SRC = Liquid solvent refined coal

Mont = Montana
N. D. = North Dakota
WCT = Wet cooling tower

FOLDOUT FRAME

Table 2.1-7

RIC VARIATIONS FOR TASK I STUDY
OPEN-CYCLE GAS TURBINE

Recuperative Cycle																		Recuperative with Organic Bottoming					
12	13	14	15	16	17	18	19	20	21	22	23	24	25	26	27	28	29	30	31	34	35	36	37
1***	331+	60	72	102	93	106	111	77	84	83	83	82	84	81	80	93	73	102	101	102	101	103	76
→	→	→	→	→	→	→	→	→	→	→	→	→	→	→	→	→	→	→	→	→	→	→	→
→	→	1800	2000	2600 →	→	→	†† →	2200 →	→	→	→	→	→	→	→	→	→	→	→	→	→	→	1800
→	→	→	→	→	8	16 →	→	8	16	12 →	→	→	→	→	→	→	→	→	→	→	→	→	→
→	→	→	→	→	→	→	→	→	→	0.8	0.9	0.95	0.85 →	→	→	→	→	→	→	→	→	→	→
→	→	→	→	→	→	→	→	→	→	→	→	→	0.0,	0.07	0.09	0.05 →	→	→	→	→	→	→	→
→	→	→	→	→	→	→	→	→	→	→	→	→	→	→	→	20	93	59 →	→	→	→	→	→
--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	10	20	10 →	→	→	→
--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	227	230	227 →	→	→	113
--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	30 →	→	50	70	30 →	→
--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	DCT 99.1 →	→	→	→	WCT 92	DCT 99.1
331	331	60	72	102	93	106	111	77	84	83	83	82	84	81	80	93	73	102	101	102	101	103	76
4.8	34.8	31.9	33.7	34.9	32.8	35.4	37.0	34.2	34.2	34.3	35.3	35.8	35.1	34.3	34.0	36.4	33.6	44.0	43.4	43.6	43.3	44.3	41.2
4.6	34.4	31.4	33.2	34.6	32.5	35.0	36.7	33.8	33.8	33.9	34.8	35.3	34.7	33.9	33.6	36.0	33.2	47.6	42.1	42.3	41.9	43.1	39.5
7.3	17.3	15.8	16.7	17.4	16.4	17.7	18.5	17.0	17.0	17.1	17.6	17.8	17.5	17.1	16.9	18.1	16.7	21.5	21.2	21.3	21.1	21.7	19.9
.82	1.82	2.00	1.89	1.82	1.93	1.79	1.71	1.86	1.86	1.85	1.80	1.78	1.81	1.85	1.87	1.74	1.89	1.47	1.49	1.49	1.50	1.46	1.59
55	54	13	13	17	16	19	19	13	15	12	16	18	15	13	12	14	14	35	35	34	34	35	32
16	162	211	187	169	171	178	169	165	175	148	191	216	174	161	154	147	190	338	342	336	337	334	427
5.3	5.1	6.7	5.9	5.4	5.4	5.6	5.4	5.2	5.6	4.7	6.1	6.8	5.5	5.1	4.9	4.7	6.0	10.7	10.8	10.6	10.7	10.6	13.5
5.8	25.8	28.3	26.7	25.7	27.3	25.3	24.2	26.3	26.3	26.2	25.5	25.1	25.6	26.2	26.4	24.7	26.8	20.8	21.1	21.0	21.2	20.6	22.5
1.6	1.6	2.3	2.0	1.7	1.9	1.7	1.6	2.3	2.1	2.1	2.1	2.1	2.1	2.2	2.2	1.9	2.4	2.6	2.6	2.6	2.6	2.5	3.5
2.7	32.5	37.3	34.6	32.8	34.6	32.6	31.2	33.8	33.9	33.0	33.6	34.1	33.2	33.4	33.5	31.2	35.2	34.1	34.5	34.2	34.4	33.7	39.5
4.7	34.5	40.0	37.0	34.9	36.8	34.8	33.2	36.0	36.2	35.0	36.1	36.8	35.5	35.6	35.7	33.2	37.7	38.1	38.5	38.2	38.4	37.7	44.6
1.4	31.3	35.6	33.1	31.4	33.2	31.3	29.8	32.4	32.5	31.7	32.1	32.4	31.8	32.1	32.2	30.0	33.6	31.6	32.0	31.8	32.0	31.3	36.3
1.1	1.0	1.3	1.2	1.1	1.1	1.1	1.1	1.0	1.1	0.9	1.2	1.4	1.1	1.0	1.0	0.9	1.2	2.1	2.2	2.1	2.1	2.1	2.7
5.2	5.2	5.7	5.3	5.1	5.5	5.1	4.8	5.3	5.3	5.2	5.1	5.0	5.1	5.2	5.3	4.9	5.4	4.2	4.2	4.2	4.2	4.1	4.5
2	2	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	2	2	2	2	2	2
980	1980	1977	1980	1985	1985	1985	1985	1980	1980	1980	1980	1980	1980	1980	1980	1980	1980	1985	1985	1985	1985	1985	1977

Table 2.1-6

SUMMARY SHEET
OPEN-CYCLE GAS TURBINE BASE CASE (RECUPERATIVE CYCLE)

CYCLE PARAMETER

Net Power Output (MWe)	83
Coal Type and Conversion Process	Illinois No. 6 HBtu
Prime Cycle	
Turbine inlet temperature (°F)	2200
Compressor pressure ratio	12
Recuperator efficiency	0.85
Recuperator pressure drop ($\Delta p/p$)	0.05
Air inlet temperature (°F)	59

MAJOR COMPONENT CHARACTERISTICS

Major Component	Unit or Module: Size (ft) (W x L (or D) x H)	Weight (lb) (x 10^6)	Cost ($ x 10^6)	Units Required	Total Cost ($ x 10^6)	$/kW Output
Gas turbine-compressor-combustor-generator	12 x 62 x 14	0.56	--	1	8.4	100.0
Recuperator	11 x 25 x 13	0.22	--	2		

PERFORMANCE AND COST

Thermodynamic efficiency (percent)	**
Powerplant efficiency (percent)	34.4
Overall energy efficiency (percent)	17.3
Plant capital cost ($ x 10^6)	14
Plant capital cost ($/kWe)	166
Cost of electricity (mills/kWh)	33.2

NATURAL RESOURCES

Coal (lb/kWh)	1.82
Water (gal/kWh)	
Total	0.026
Cooling	0
Processing	0
Makeup	0
NO_x suppression	0.026
Stack gas cleanup	0
Land (acres/100 MWe)	3.01

ENVIRONMENTAL INTRUSION

	Lb/10^6-Btu Input	Lb/kWh Output
SO_2	0.009	0.18×10^{-3}
NO_x	0.2	3.93×10^{-3}
HC	0	0
CO	0	0
Particulates	0	0

	Btu/kWh
Heat to water	0
Heat, total rejected	6507
Wastes	--

CAPITAL COST DISTRIBUTIONS FOR OPEN-CYCLE GAS TURBINE

	CASE NO.	1	2	3	4	5	6	7	8	9	10
MAJOR COMPONENTS											
PRIME CYCLE											
GAS TURB-COMP-COMB-GEN-RECUP	MM$	6.3	9.6	9.6	11.2	13.9	8.4	8.4	8.4	9.4	9.4
BOTTOMING CYCLE											
ORGANIC TURB-GEN	MM$	0.	0.	0.	0.	0.	0.	0.	0.	0.	0.
ORGANIC BOILER	MM$	0.	0.	0.	0.	0.	0.	0.	0.	0.	0.
ORGANIC RECUPERATOR	MM$	0.	0.	0.	0.	0.	0.	0.	0.	0.	0.
ORGANIC CONDENSER	MM$	0.	0.	0.	0.	0.	0.	0.	0.	0.	0.
SUB-TOTAL OF MAJOR COMPONENTS	MM$	6.3	9.6	9.6	11.2	13.9	8.4	8.4	8.4	9.4	9.4
BALANCE OF PLANT											
COOLING TOWER	MM$	0.	0.	0.	0.	0.	0.	0.	0.	0.	0.
ALL OTHER	MM$	1.6	1.6	1.6	1.6	1.6	1.6	1.6	1.6	1.6	1.6
SITE LABOR	MM$	0.3	0.3	0.3	0.3	0.3	0.3	0.3	0.3	0.3	0.3
SUB-TOTAL OF BALANCE OF PLANT	MM$	1.8	1.8	1.8	1.8	1.8	1.9	1.9	1.9	1.9	1.9
CONTINGENCY	MM$	1.6	2.3	2.3	2.6	3.1	2.1	2.1	2.1	2.3	2.3
ESCALATION COSTS	MM$	0.6	0.9	0.9	1.0	1.2	0.8	0.8	0.8	0.9	0.9
INTEREST DURING CONSTRUCTION	MM$	0.5	0.7	0.7	0.8	1.0	0.7	0.7	0.7	0.7	0.7
TOTAL CAPITAL COST	MM$	11.0	15.3	15.4	17.5	21.1	13.8	13.8	13.8	15.1	15.1
MAJOR COMPONENTS COST	$/KWE	72.5	86.5	84.0	86.0	65.6	101.7	101.7	101.7	117.5	119.0
BALANCE OF PLANT	$/KWE	20.9	16.5	15.9	14.0	8.6	22.8	22.8	22.8	23.6	23.7
CONTINGENCY	$/KWE	18.7	20.6	20.0	20.0	14.8	24.9	24.9	24.9	28.2	28.3
ESCALATION COSTS	$/KWE	7.3	8.0	7.8	7.8	5.8	9.7	9.7	9.7	11.0	11.1
INTEREST DURING CONSTRUCTION	$/KWE	6.0	6.6	6.4	6.4	4.7	8.0	8.0	8.0	9.0	9.1
TOTAL CAPITAL COST	$/KWE	125.3	138.2	134.1	134.2	99.5	167.0	167.0	167.0	189.2	190.2

Table 2.1-8 (Page 2 of 4)

CAPITAL COST DISTRIBUTIONS FOR OPEN-CYCLE GAS TURBINE

	CASE NO.	11	12	13	14	15	16	17	18	19	20
MAJOR COMPONENTS											
PRIME CYCLE											
GAS TURB-COMP-COMB-GEN-RECUP	MM$	3.5	33.6	32.6	7.6	8.1	11.1	10.0	12.2	12.1	7.6
BOTTOMING CYCLE											
ORGANIC TURB-GEN	MM$	0.	0.	0.	0.	0.	0.	0.	0.	0.	0.
ORGANIC BOILER	MM$	0.	0.	0.	0.	0.	0.	0.	0.	0.	0.
ORGANIC RECUPERATOR	MM$	0.	0.	0.	0.	0.	0.	0.	0.	0.	0.
ORGANIC CONDENSER	MM$	0.	0.	0.	0.	0.	0.	0.	0.	0.	0.
SUB-TOTAL OF MAJOR COMPONENTS	MM$	3.5	33.6	32.6	7.6	8.1	11.1	10.0	12.2	12.1	7.6
BALANCE OF PLANT											
COOLING TOWER	MM$	0.	0.	0.	0.	0.	0.	0.	0.	0.	0.
ALL OTHER	MM$	0.3	3.5	3.5	1.6	1.6	1.6	1.6	1.6	1.6	1.6
SITE LABOR	MM$	0.1	1.1	1.1	0.3	0.3	0.3	0.3	0.3	0.3	0.3
SUB-TOTAL OF BALANCE OF PLANT	MM$	0.4	4.6	4.6	1.9	1.9	1.9	1.9	1.9	1.9	1.9
CONTINGENCY	MM$	0.[illegible]	7.6	7.4	1.9	2.0	2.6	2.4	2.8	2.8	1.9
ESCALATION COSTS	MM$	0.3	5.1	5.0	0.7	0.8	1.0	0.9	1.1	1.1	0.7
INTEREST DURING CONSTRUCTION	MM$	0.3	4.2	4.1	0.6	0.6	0.8	0.8	0.9	0.9	0.6
TOTAL CAPITAL COST	MM$	5.3	55.2	53.7	12.7	13.4	17.4	16.0	18.8	18.8	12.7
MAJOR COMPONENTS COST	$/KWE	142.8	101.7	98.6	126.5	113.4	108.0	107.3	115.1	109.6	98.9
BALANCE OF PLANT	$/KWE	17.6	13.9	13.9	31.3	26.3	13.4	20.2	17.8	17.0	24.5
CONTINGENCY	$/KWE	32.1	23.1	22.5	31.6	27.9	25.3	25.5	26.6	25.3	24.7
ESCALATION COSTS	$/KWE	12.5	15.5	15.1	12.3	10.9	9.4	9.9	10.4	9.9	9.6
INTEREST DURING CONSTRUCTION	$/KWE	10.3	12.7	12.4	10.1	8.9	8.1	8.1	8.5	8.1	7.9
TOTAL CAPITAL COST	$/KWE	215.3	166.9	162.5	211.7	187.4	169.6	171.0	178.4	169.9	165.5

Table 2.1-8 (Page 3 of 4)

CAPITAL COST DISTRIBUTIONS FOR OPEN-CYCLE GAS TURBINE

	CASE NO.	21	22	23	24	25	26	27	28	29	30
MAJOR COMPONENTS											
PRIME CYCLE											
GAS TURB-COMP-COMB-GEN-RECUP	MM$	9.1	7.3	9.9	11.4	9.0	7.9	7.3	8.4	8.4	8.4
BOTTOMING CYCLE											
ORGANIC TURB-GEN	MM$	0.	0.	0.	0.	0.	0.	0.	0.	0.	1.4
ORGANIC BOILER	MM$	0.	0.	0.	0.	0.	0.	0.	0.	0.	1.0
ORGANIC RECUPERATOR	MM$	0.	0.	0.	0.	0.	0.	0.	0.	0.	0.5
ORGANIC CONDENSER	MM$	0.	0.	0.	0.	0.	0.	0.	0.	0.	1.1
SUB-TOTAL OF MAJOR COMPONENTS	MM$	9.1	7.3	9.9	11.4	9.0	7.9	7.3	8.4	8.4	12.4
BALANCE OF PLANT											
COOLING TOWER	MM$	0.	0.	0.	0.	0.	0.	0.	0.	0.	0.5
ALL OTHER	MM$	1.6	1.6	1.6	1.6	1.6	1.6	1.6	1.6	1.6	7.7
SITE LABOR	MM$	0.3	0.3	0.3	0.3	0.3	0.3	0.3	0.3	0.3	3.4
SUB-TOTAL OF BALANCE OF PLANT	MM$	1.9	1.9	1.9	1.9	1.9	1.9	1.9	1.9	1.9	11.6
CONTINGENCY	MM$	2.2	1.8	2.4	2.7	2.2	2.0	1.8	2.1	2.1	4.8
ESCALATION COSTS	MM$	0.9	0.7	0.9	1.0	0.8	0.8	0.7	0.8	0.8	3.2
INTEREST DURING CONSTRUCTION	MM$	0.7	0.6	0.8	0.8	0.7	0.6	0.6	0.7	0.7	2.6
TOTAL CAPITAL COST	MM$	14.7	12.3	15.8	17.8	14.6	13.1	12.4	13.8	13.8	34.6
MAJOR COMPONENTS COST	$/KWE	108.5	87.7	119.8	138.4	107.3	97.0	91.7	90.1	115.7	121.0
BALANCE OF PLANT	$/KWE	22.4	22.8	22.8	22.8	22.5	23.2	23.5	20.2	25.9	113.3
CONTINGENCY	$/KWE	26.2	22.1	28.5	32.3	26.0	24.0	23.0	22.1	28.3	46.9
ESCALATION COSTS	$/KWE	10.2	8.6	11.1	12.6	10.1	9.4	9.0	8.6	11.0	31.4
INTEREST DURING CONSTRUCTION	$/KWE	8.4	7.1	9.1	10.3	8.3	7.7	7.4	7.0	9.1	25.8
TOTAL CAPITAL COST	$/KWE	175.8	148.3	191.4	216.4	174.2	161.2	154.6	147.9	190.1	338.4

Table 2.1-8 (Page 4 of 4)

CAPITAL COST DISTRIBUTIONS FOR OPEN-CYCLE GAS TURBINE

	CASE NO.	31	34	35	36	37
MAJOR COMPONENTS						
PRIME CYCLE						
GAS TURB-COMP-COMB-GEN-RECUP.	MM$	8.4	8.4	8.4	8.4	7.6
BOTTOMING CYCLE						
ORGANIC TURB-GEN	MM$	1.4	1.3	1.3	1.4	1.2
ORGANIC BOILER	MM$	1.0	0.8	0.6	1.1	0.8
ORGANIC RECUPERATOR	MM$	0.5	0.5	0.5	0.5	0.4
ORGANIC CONDENSER	MM$	1.1	1.1	1.1	1.1	0.9
SUB-TOTAL OF MAJOR COMPONENTS	MM$	12.4	12.1	11.9	12.5	10.9
BALANCE OF PLANT						
COOLING TOWER	MM$	0.5	0.5	0.5	0.2	0.5
ALL OTHER	MM$	7.7	7.7	7.7	8.1	7.7
SITE LABOR	MM$	3.4	3.4	3.4	3.2	3.4
SUB-TOTAL OF BALANCE OF PLANT	MM$	11.6	11.6	11.6	11.5	11.6
CONTINGENCY	MM$	4.8	4.7	4.7	4.8	4.5
ESCALATION COSTS	MM$	3.2	3.2	3.2	3.2	3.0
INTEREST DURING CONSTRUCTION	MM$	2.6	2.6	2.6	2.6	2.5
TOTAL CAPITAL COST	MM$	34.6	34.2	34.0	34.7	32.4
MAJOR COMPONENTS COST	$/KWE	122.4	118.9	118.4	120.6	143.3
BALANCE OF PLANT	$/KWE	114.6	114.2	115.1	111.3	152.9
CONTINGENCY	$/KWE	47.4	46.6	46.7	46.4	59.3
ESCALATION COSTS	$/KWE	31.8	31.3	31.3	31.1	39.7
INTEREST DURING CONSTRUCTION	$/KWE	26.1	25.7	25.7	25.6	32.7
TOTAL CAPITAL COST	$/KWE	342.3	336.7	337.2	334.9	427.9

DISCUSSION OF RESULTS

Simple Cycle Gas Turbine

The air-cooled Cases 1, 2, and 4 for the simple cycle machine show that efficiency peaks at 2600 F (1700 K) and a pressure ratio of 16 and then decreases at higher temperature as a result of the accelerated use of air for cooling. The cost of electricity is minimum there also. These units are very sensitive to fuel cost and to efficiency, with small sensitivity to capital cost. Use of ceramic stationary parts improves efficiency marginally but not dramatically. By contrast, the very complex water-cooled turbine of Case 5 produces power at the same high efficiency, but with plant cost at $99/kW. This case indicates the advantage that might accrue to such advanced developments.

Recuperative Gas Turbines

A comparison of selected cases using conditions for Case 6 of 2200 F (1480 K) and 12 pressure ratio in Table 2.1-9 shows that the simple cycle was least costly, the recuperative cycle was most economic for the generation of power, and the recuperative cycle with organic bottoming was the most costly power plant.

Table 2.1-9

CYCLE COMPARISONS FOR 2200 F, 12 PRESSURE RATIO

Configuration (case)	Plant Efficiency (%)	Cost $/kW	Cost mills/kWh
Simple cycle (1)	30	125	36
Recuperative cycle (6)	34	166	33
Recuperative with (30) organic bottoming	43	388	34

The efficiency improves with recuperation, and even more dramatically to 43 percent with organic bottoming for net plant power divided by the higher heating value of the high-Btu gas consumed. The more than doubling of the capital cost per kilowatt for the organic bottomed plant results in an overall cost for electricity nearly equal to that for the recuperative machine.

The capital cost distribution, Table 2.1-8, clearly shows the source of the cost disadvantage of Case 30 as compared with Case 6. Table 2.1-10 shows that the power increment costs $1040/kW for the addition of organic bottoming to the recuperative gas turbine.

DISCUSSION OF RESULTS

Simple Cycle Gas Turbine

The air-cooled Cases 1, 2, and 4 for the simple cycle machine show that efficiency peaks at 2600 F (1700 K) and a pressure ratio of 16 and then decreases at higher temperature as a result of the accelerated use of air for cooling. The cost of electricity is minimum there also. These units are very sensitive to fuel cost and to efficiency, with small sensitivity to capital cost. Use of ceramic stationary parts improves efficiency marginally but not dramatically. By contrast, the very complex water-cooled turbine of Case 5 produces power at the same high efficiency, but with plant cost at $99/kW. This case indicates the advantage that might accrue to such advanced developments.

Recuperative Gas Turbines

A comparison of selected cases using conditions for Case 6 of 2200 F (1480 K) and 12 pressure ratio in Table 2.1-9 shows that the simple cycle was least costly, the recuperative cycle was most economic for the generation of power, and the recuperative cycle with organic bottoming was the most costly power plant.

Table 2.1-9

CYCLE COMPARISONS FOR 2200 F, 12 PRESSURE RATIO

Configuration (case)	Plant Efficiency (%)	Cost $/kW	Cost mills/kWh
Simple cycle (1)	30	125	36
Recuperative cycle (6)	34	166	33
Recuperative with (30) organic bottoming	43	388	34

The efficiency improves with recuperation, and even more dramatically to 43 percent with organic bottoming for net plant power divided by the higher heating value of the high-Btu gas consumed. The more than doubling of the capital cost per kilowatt for the organic bottomed plant results in an overall cost for electricity nearly equal to that for the recuperative machine.

The capital cost distribution, Table 2.1-8, clearly shows the source of the cost disadvantage of Case 30 as compared with Case 6. Table 2.1-10 shows that the power increment costs $1040/kW for the addition of organic bottoming to the recuperative gas turbine.

Table 2.1-10

INCREMENTAL COST FOR ORGANIC BOTTOMING CYCLES

Item	Δ$/ΔkW	
Turbine generator	70	
Organic boiler	50	
Organic recuperator	25	
Organic condenser	55	
Major components		200
Cooling tower	25	
Labor at site	155	
Other balance of plant	305	
Balance of plant		485
Contingency, escalation, interest		355
Total		1040

The organic bottoming cycle is a viable route to enhanced efficiency, but contributes disproportionate costs relative to the increment in power realized. Since balance-of-plant costs for field assembly of the elements of the organic fluid unit are large capital adders, it was concluded that efforts should be directed toward unitized and factory assembled organic bottoming modules in place of the field erection and integration of the 25 MW bottoming system.

The most economic power generation for the assumptions of this study resulted from use of the most advanced gas turbine conditions when high-Btu gas was the fuel. These were the 3000 F (1920 K), 16 pressure ratio water-cooled simple cycle gas turbine and the 2600 F (1700 K), 16 pressure ratio air-cooled recuperative gas turbine using ceramic stationary parts in the hot gas path. Table 2.1-11 presents these results along with values at standard 2200 F (1480 K), 12 pressure ratio for alternate fuels.

Table 2.1-11

LOWEST COST FOR ELECTRICITY

Configuration	Temperature (^{o}F)/ Pressure Ratio	Cost-Various Fuels (mills/kWh) LBtu	COED	SRC
Simple cycle	3000/16	32	-	-
Recuperative cycle	2600/16	31	-	-
Recuperative cycle	2200/12	33	32	26

The influence of firing temperature and pressure ratio is shown in Table 2.1-12. The major effect is due to increased temperature with addition of ceramic stationary parts producing further improvement.

Table 2.1-12

INLET TEMPERATURE EFFECTS

Temperature (°F)	Pressure Ratio	Cost (Mills/kWh)
2200	12	33.2
2600	12	32.8
2600	16	32.6
2600, ceramic parts	16	31.2

The advantage of larger installation size was examined in Cases 12 and 13 with four times the capacity of Base Case 6. Despite the penalties of escalation and interest during construction due to an increase of one year in construction time these plants showed a marginal reduction in capital cost per kilowatt. The uniform level of operating cost resulted in a reduction of the operating and maintenance item of generation cost. As a result the larger installations show a 1.5 percent advantage in generation cost.

Cases 22 through 27 examined the economic impact of recuperator effectiveness and pressure losses. The results show minimum electricity generation costs with 80 to 85 percent effectiveness and 3 to 5 percent pressure loss. These values are typical of current commercial recuperators for gas turbines.

Table 2.1-13 presents the accounting for the generated power and for auxiliary consumption leading to the net station output in each case.

Observations

All gas turbine cycles required water to limit thermal NO_X generation to the emission standards. The liquid solvent refined coal fuel (SRC) has so much fuel-bound nitrogen that the NO_X exceeds emission standards under any firing conditions. The water consumption is minimal by any power production comparison.

The capital costs are very low as a result of modular delivery on site that minimizes field assembly labor and total construction time.

The simple and recuperative gas turbines are conducive to unattended operation.

Table 2.1-13

POWER OUTPUT AND AUXILIARY POWER DEMAND FOR BASE CASE AND PARAMETRIC VARIATIONS: OPEN-CYCLE GAS TURBINE

CASE NO.		1	2	3	4	5	6	7	8	9	10
PRIME CYCLE POWER OUTPUT	MW	88.9	112.2	116.2	132.1	214.0	84.1	84.1	84.1	81.3	80.9
BOTTOMING CYCLE POWER OUTPUT	MW	0.	0.	0.	0.	0.	0.	0.	0.	[illegible]	0.
FURNACE POWER OUTPUT	MW	0.	0.	0.	0.	0.	0.	0.	0.	0.	0.
BALANCE OF PLANT AUX. POWER REQ'D.	MW	1.0	1.0	1.0	1.0	1.0	1.0	1.0	1.0	1.0	1.0
FURNACE AUX. POWER REQ'D.	MW	0.	0.	0.	0.	0.	0.	0.	0.	0.	0.
TRANSFORMER LOSSES	MW	0.4	0.6	0.6	0.7	1.1	0.4	0.4	0.4	0.4	0.4
NET STATION OUTPUT	MW	87.5	110.7	114.6	130.5	211.9	82.7	82.7	82.7	79.9	79.5

CASE NO.		11	12	13	14	15	16	17	18	19	20
PRIME CYCLE POWER OUTPUT	MW	25.0	336.4	336.4	61.4	73.1	103.9	94.8	107.1	112.3	78.4
BOTTOMING CYCLE POWER OUTPUT	MW	0.	0.	0.	0.	0.	0.	0.	0.	0.	0.
FURNACE POWER OUTPUT	MW	0.	0.	0.	0.	0.	0.	0.	0.	0.	0.
BALANCE OF PLANT AUX. POWER REQ'D.	MW	0.2	4.0	4.0	1.0	1.0	1.0	1.0	1.0	1.0	1.0
FURNACE AUX. POWER REQ'D.	MW	0.	0.	0.	0.	0.	0.	0.	0.	0.	0.
TRANSFORMER LOSSES	MW	0.2	1.7	1.7	0.3	0.4	0.5	0.5	0.5	0.6	0.4
NET STATION OUTPUT	MW	24.6	330.7	330.7	60.1	71.7	102.4	93.3	105.6	110.8	77.0

CASE NO.		21	22	23	24	25	26	27	28	29	30
PRIME CYCLE POWER OUTPUT	MW	85.3	84.2	84.0	83.9	85.0	82.6	81.4	94.8	74.0	82.7
BOTTOMING CYCLE POWER OUTPUT	MW	0.	0.	0.	0.	0.	0.	0.	0.	0.	23.5
FURNACE POWER OUTPUT	MW	0.	0.	0.	0.	0.	0.	0.	0.	0.	0.
BALANCE OF PLANT AUX. POWER REQ'D.	MW	1.0	1.0	1.0	1.0	1.0	1.0	1.0	1.0	1.0	3.3
FURNACE AUX. POWER REQ'D.	MW	0.	0.	0.	0.	0.	0.	0.	0.	0.	0.
TRANSFORMER LOSSES	MW	0.4	0.4	0.4	0.4	0.4	0.4	0.4	0.5	0.4	0.6
NET STATION OUTPUT	MW	83.9	82.8	82.6	82.5	83.6	81.2	80.0	93.3	72.6	102.4

CASE NO.		31	34	35	36	37
PRIME CYCLE POWER OUTPUT	MW	81.4	82.7	82.7	82.7	60.4
BOTTOMING CYCLE POWER OUTPUT	MW	23.7	22.7	21.9	24.2	19.1
FURNACE POWER OUTPUT	MW	0.	0.	0.	0.	0.
BALANCE OF PLANT AUX. POWER REQ'D.	MW	3.3	3.3	3.3	2.9	3.3
FURNACE AUX. POWER REQ'D.	MW	0.	0.	0.	0.	0.
TRANSFORMER LOSSES	MW	0.6	0.6	0.6	0.6	0.4
NET STATION OUTPUT	MW	101.1	101.6	100.8	103.5	75.8

The addition of organic bottoming enhances efficiency and reduces fuel consumption; the impact on capital, on construction time, on on-site labor, on ease of operation, and on economics are all adverse.

RECOMMENDED CASES

Case 19 with ceramic nozzles 2600 F (1700 K), 16 pressure ratio had the highest power plant efficiency of all unbottomed cases and is recommended for further study. The on-site efficiency was 37 percent with high-Btu fuel resulting in 31.2 mills/kWh electricity at a 65 percent capacity factor.

An alternative with even higher on-site efficiency would be the organic bottomed recuperative cycle for 2200 F (1480 K), 12 pressure ratio with 44 percent on-site efficiency resulting in 34.1 mills/kWh electricity at 65 percent capacity factor.

2.2 OPEN-CYCLE GAS TURBINE COMBINED CYCLE–AIR COOLED

DESCRIPTION OF CYCLE

The schematic of this cycle is presented in Figure 2.2-1 showing the gas turbines integrated with the fixed bed type low-Btu gasifier, with a heat recovery steam generator (HRSG), and with the bottoming steam turbine which is also integrated with the gasifier. Approximately 11 percent of the compressed air from the four gas turbines flows to the booster compressor and then to the gasifiers. The gasifier water jacket, the low pressure drum of the HRSG, and extraction of steam from the main steam turbine provides steam for gasification of coal. The low-Btu fuel gas is fired in the gas turbines. The gas turbine exhaust produces high-pressure steam in the HRSG as well as low-pressure steam for gasification. The high-pressure steam expands through the first turbine section. Approximately 30 percent of the turbine steam is extracted to fulfill the gasifier steam demand and to power the 11 MW booster air compressor. The steam is condensed using coolant from the wet cooling towers. Makeup water is provided continuously and is heated and deaerated along with the condensate. 448 MW of power are generated by four gas turbines and 150 MW by the one steam turbine. Power is consumed by the wet cooling towers, the condensate and boiler feedpumps, the gasifier drive and gas cleanup drives and other plant auxiliaries. There is no neat way to subdivide and account for power generation to estimate a thermodynamic efficiency in such a highly interdependent system. Accordingly the basis for efficiency is the net plant output divided by the higher heating value of the coal supplied.

Variations on the HRSG, such as use of one drum and provision for reheating steam and for firing low-Btu gas at the HRSG, are detailed in Section 4, "Bottoming Cycles."

For firing other fuels such as intermediate-Btu gas, high-Btu gas, and liquid fuels, and for firing low-Btu gas delivered to the site, the schematic of Figure 2.2-1 is greatly simplified. Only the gas turbines, HRSG, bottoming steam turbine, condenser, and cooling towers remain. A small steam supply to the gas turbine would be added for the high-Btu gas and liquid fuel fired turbines for NO_X suppression. The intermediate- and low-Btu gas power plants require no water or steam for NO_X suppression. Intermediate-Btu power plants could operate without continuous water demand if they used dry cooling towers.

ANALYTICAL PROCEDURES AND ASSUMPTIONS

All air-cooled gas turbines were of the type described in Section 2.1 and had an inlet airflow of 570 lb/s (258.5 kg/s). The single exception was Case 2 wherein a single 1800 RPM gas turbine replaced the assigned four gas turbines. The low-Btu gas results in greater than normal mass flow through the turbine portion of the gas turbine. The analytical procedures and assumptions remain the same as those described in Section 2.1. For the

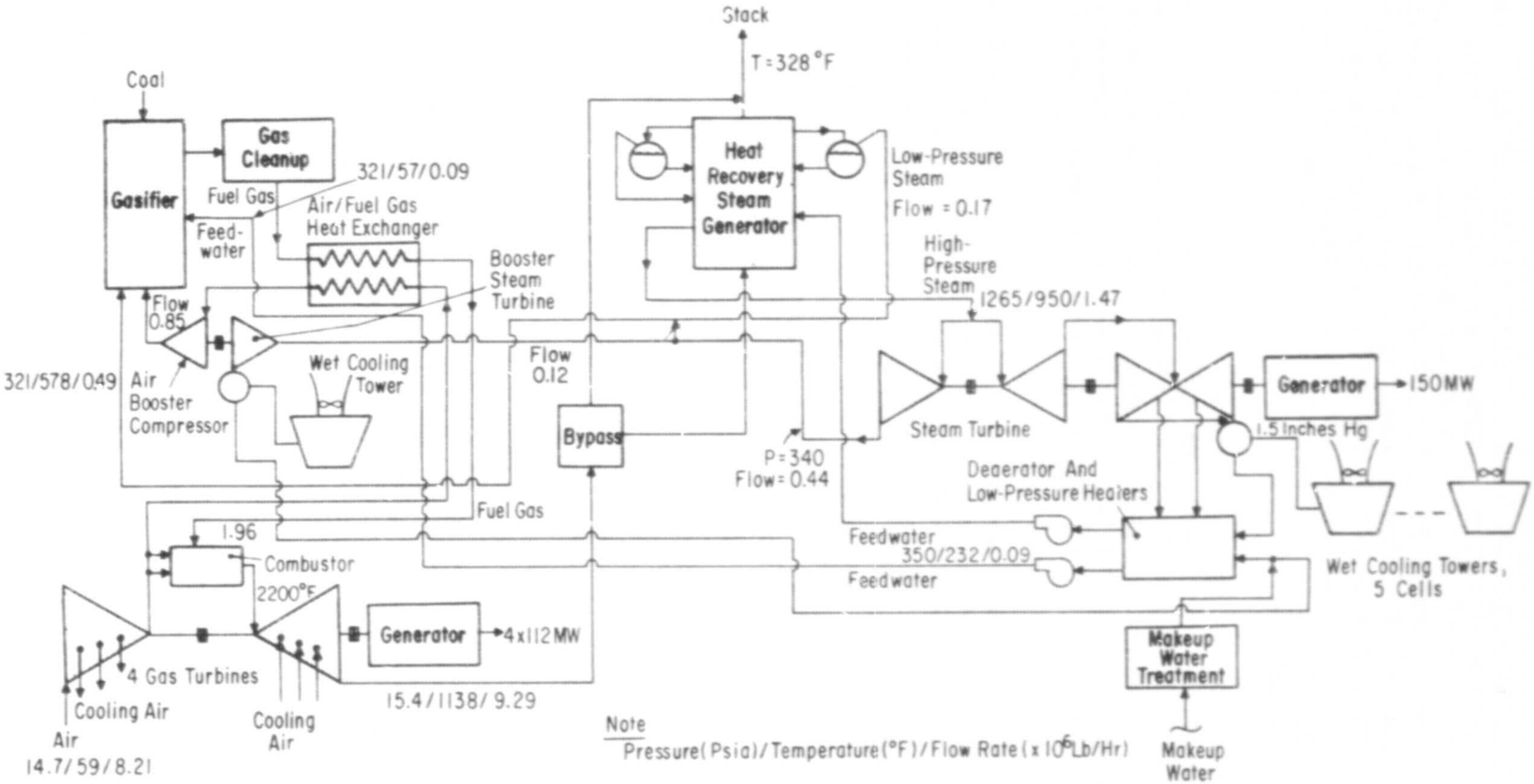

Figure 2.2-1. Open-Cycle Gas Turbine Combined Cycle—Air Cooled

HRSG an additional 15 inches of water pressure loss is added to the gas turbine exhaust to make a total of 20 inches. The HRSG pinch point was specified with the result that the final stack temperature relates to the initial gas turbine exhaust temperature and the energy extracted in the several HRSG elements as detailed in Section 4. The feedwater temperature from the deaerating feedheater was 232 F (384 K). Whenever steam turbine reheat was used, the low-pressure drum supplying gasifier steam was eliminated. Section 4 presents the medium steam turbine bottoming cycle in greater detail.

The steam-driven booster compressor supplying air to the gasifier was assigned 75 percent efficiency. The fuel gas to air recuperator was 80 percent effective with a pressure loss of 2 percent on each side. Three percent pressure losses were assigned from the compressor to the recuperator, from the recuperator to the booster compressor, and from the booster compressor to the gasifier.

The condenser for the steam turbine would be serviced by cooling towers. A condensing pressure of 1.5 in. Hga (5 kN/m^2) was assigned for use with wet cooling towers for all but two cases. For these two cases dry cooling towers were considered which produced 1.9 in. Hga (6.4 kN/m^2), for a 40 F (22 K) difference between ambient air and condensing temperature; and 3.45 in. Hga (11.7 kN/m^2) for a 60 F (33 K) difference. Only this last case required a high back pressure steam turbine design rather than a conventional design.

The low-Btu gasifier has been described in Section 8. The discharge pressure assigned for the clean fuel gas was 227 psia (1.6 MN/m^2) for gas turbine compressor pressure ratios of 8 and 12, and 303 psia (2.1 MN/m^2) for pressure ratios of 16 and 20. The latter would require a booster compressor to deliver the fuel gas to the gas turbine combustor.

DESIGN AND COST BASIS

The HRSG is a component not previously considered. The heat transfer area was evaluated on a row by row basis to assure that pinch points of minimum temperature difference were honored. The drums were proportioned to produce a fixed volume rate of steam release. The cost was then composed of the element costs and the assembly costs including the appropriate controls.

The steam turbine-generator cost basis was identical to that described for large steam turbines in Section 2.6 and need not be repeated. There are explicit cost additions allocated for the extraction of large steam flows as required for these steam turbines.

The design and cost basis for the gasifiers are described in Section 8. The balance-of-plant costs as reported by the architect-

FOLDOUT FRAME /

PARAMETRI
OPEN-CYCLE GA

Parameters	Case 1*	2	3	4	5	6	7	8	9	10	11	
Power Output (MWe)	576	576 **	599	592	514	520	517	476	457	461	285	11
Coal and Conversion Process	Ill. #6 LBtu	→	N.D. LBtu	Mont LBtu	Ill. #6 IBtu	N.D. IBtu •	Mont IBtu	Ill. #6 HBtu	Ill. #6 SRC	Ill. #6 Coed	Ill. #6 LBtu	
Prime Cycle												
Turbine inlet temperature (°F)	2200											
Compressor pressure ratio	12											
Gas path (Δp in. H_2O)	20											
Bottoming Cycle												
Turbine inlet temperature (°F)	950											
Turbine inlet pressure (psi)	1250											
Condenser (in. Hga)	1.5											
Heat Exchanger												
ΔT gas to superheater (minimum)	50											
ΔT boiler pinch point (nominal)	30											
Supplemental firing (1400°F)	0											
Heat Rejection	WCT											
Actual Powerplant Output (MWe)	576	576	599	592	514	520	517	476	457	461	285	11
Thermodynamic Efficiency (percent)	0.	0.	0.	0.	46.9	46.3	46.4	42.6	45.0	44.4	0.	0
Powerplant Efficiency (percent)	35.6	35.6	35.8	37.6	45.6	45.1	45.2	41.4	43.6	43.1	35.3	35
Overall Energy Efficiency (percent)	35.6	35.6	35.8	37.6	31.8	32.0	32.1	20.9	34.0	24.1	35.3	35
Coal Consumption (lb/kWh)	0.89	0.89	1.39	1.01	0.99	1.55	1.19	1.52	0.93	1.31	0.70	0.
Plant Capital Cost ($ million)	259	258	289	272	131	131	131	107	125	127	142	5
Plant Capital Cost ($/kWe)	450	448	482	459	255	252	253	225	274	276	498	4
Cost of Electricity, Capacity Factor = 0.65												
Capital (mills/kWh)	14.2	14.2	15.2	14.5	8.1	8.0	8.0	7.1	8.7	8.7	15.8	14
Fuel (mills/kWh)	8.2	8.2	8.1	7.7	15.0	15.1	15.1	21.4	14.1	20.6	8.2	8
Maintenance and operating (mills/kWh)	3.1	3.1	3.2	3.1	1.7	1.7	1.7	1.8	2.0	2.0	3.8	2
Total (mills/kWh)	25.5	25.5	26.6	25.4	24.8	24.8	24.8	30.4	24.8	31.4	27.7	25
Sensitivity												
Capacity factor = 0.50 (total mills/kWh)	30.8	30.7	32.1	30.7	27.7	27.7	27.7	33.1	28.0	34.6	33.6	30
Capacity factor = 0.80 (total mills/kWh)	22.3	22.2	23.1	22.1	22.9	23.0	23.0	28.7	22.8	29.3	24.1	22
Capital Δ = 20 percent (Δmills/kWh)	2.8	2.8	3.0	2.9	1.6	1.6	1.6	1.4	1.7	1.7	3.2	2
Fuel Δ = 20 percent (Δmills/kWh)	1.6	1.6	1.6	1.5	3.0	3.0	3.0	4.3	2.8	4.1	1.6	1
Estimated Time for Construction (years)	3	3	3	3	2	2	2	2	3	3	3	
Estimated Date of 1st Commercial Service (year)	1982	1982	1982	1982	1982	1982	1982	1982	1982	1982	1982	19

*Base case -- Four gas turbines and one steam turbine, except cases 2, 11, and 12
**1800 rpm
+ Intercooled compressor
# Ceramic nozzle

DCT = Dry cooling tower
HBtu = High Btu
IBtu = Intermediate Btu
Ill. = Illinois
LBtu = Low Btu

Mont = Montana
N. D. = North Dakota
SRC = Solvent refined coal
WCT = Wet cooling tower

Table 2.2-1

VARIATIONS FOR TASK I STUDY
S TURBINE COMBINED CYCLE—AIR COOLED

2	13	14	15	16	17	18	19	20	21	22	23	24	25	26	27	30	31	32	33	34	35	36	37
55	481	677	771	687	752	770	510	677	594	579	572	574	577	571	579	567	567	647	566	662	770	716	575
→	→	→	→	→	→	→	→	→	→	→	→	→	→	→	→	→	→	→	→	→	→	→	→
→	2000	2400	2600 →	→	→	→	2200	2600 →	†	2200 →	→	→	→	→	→	→	→	→	→	→	2600 →	††	2200
→	→	→	→	16	8 →	→	16	20† →	→	12 →	→	→	→	→	→	→	→	→	→	→	→	16	12
→	→	→	→	→	→	→	→	→	→	10	30	20 →	→	→	→	→	→	→	→	→	→	→	→
→	→	→	→	→	→	950/950	950 →	→	→	→	→	900	1000	950 →	→	→	→	→	→	950/950	1000	950 →	→
→	→	→	→	→	→	1800	1250 →	→	→	→	→	→	→	1000	1500	1250 →	→	→	→	1800	1450	1250 →	→
→	→	→	→	→	→	→	→	→	→	→	→	→	→	→	→	→	→	→	3.45	1.5 →	→	→	1.9
→	→	→	→	→	→	→	→	→	→	→	→	→	→	→	→	→	→	→	→	→	→	→	→
→	→	→	→	→	→	→	→	→	→	→	→	→	→	→	→	50	70	30 →	→	→	→	→	→
→	→	→	→	→	→	→	→	→	→	→	→	→	→	→	→	→	→	Fired	0	Fired	0 →	→	→
→	→	→	→	→	→	→	→	→	→	→	→	→	→	→	→	→	→	→	DCT	WCT →	→	→	DCT
5	481	677	771	687	752	770	510	677	594	579	572	574	577	571	579	567	567	647	566	662	779	716	575
	0.	0.	0.	0.	0.	0.	0.	0.	0.	0.	0.	0.	0.	0.	0.	0.	0.	0.	0.	0.	0.	0.	0.
7	35.0	36.2	36.4	37.2	33.6	34.3	35.9	34.6	37.0	35.8	35.4	35.5	35.7	35.3	35.8	35.1	35.0	33.7	35.0	34.5	36.7	37.5	35.5
7	35.0	36.2	36.4	37.2	33.6	34.3	35.9	34.6	37.0	35.8	35.4	35.5	35.7	35.3	35.8	35.1	35.0	33.7	35.0	34.5	36.7	37.5	35.5
9	0.90	0.87	0.87	0.85	0.94	0.92	0.88	0.91	0.85	0.88	0.89	0.89	0.89	0.90	0.88	0.90	0.90	0.94	0.90	0.92	0.86	0.84	0.89
3	241	284	308	285	316	312	245	310	287	254	252	259	260	259	261	257	257	284	263	283	308	300	269
1	501	420	398	415	419	404	479	457	482	438	430	451	450	453	450	453	453	437	463	427	395	418	468
6	15.9	13.3	12.6	13.1	13.3	12.8	15.1	14.5	15.3	13.9	13.9	14.3	14.2	14.4	16.2	14.3	14.3	13.8	14.7	13.5	12.5	13.2	14.8
1	8.3	8.0	8.0	7.8	8.6	8.4	8.1	8.4	7.8	8.1	8.2	8.2	8.1	8.2	8.1	8.3	8.3	8.6	8.3	8.4	7.9	7.7	8.2
7	3.3	2.8	2.6	2.7	2.7	2.7	3.3	2.9	3.0	3.1	3.2	3.1	3.1	3.2	3.1	3.2	3.2	3.0	3.2	2.8	2.5	2.6	3.1
4	27.5	24.1	23.2	23.6	24.6	24.0	26.5	25.7	26.1	25.1	25.3	25.6	25.5	25.7	25.5	25.8	25.8	25.4	26.1	24.8	22.9	23.5	26.1
6	33.2	28.9	27.7	28.3	29.4	28.6	32.1	30.9	31.6	30.2	30.4	30.8	30.7	31.0	30.7	31.0	31.1	30.4	31.5	29.7	27.4	28.3	31.5
2	23.9	21.1	20.3	20.6	21.6	21.1	23.1	22.5	22.7	21.9	22.1	22.3	22.3	22.5	22.2	22.5	22.5	22.2	22.8	21.7	20.1	20.6	22.7
9	3.2	2.7	2.5	2.6	2.7	2.6	3.0	2.9	3.1	2.8	2.8	2.9	2.8	2.9	2.8	2.9	2.9	2.8	2.9	2.7	2.5	2.6	3.0
6	1.7	1.6	1.6	1.6	1.7	1.7	1.6	1.7	1.6	1.6	1.6	1.6	1.6	1.6	1.6	1.7	1.7	1.7	1.7	1.7	1.6	1.5	1.6
4	3	3	3	3	3	3	3	3	3	3	3	3	3	3	3	3	3	3	3	3	3	3	3
2	1070	1984	1984	1984	1984	1984	1982	1984	1984	1982	1982	1982	1982	1982	1982	1982	1982	1982	1982	1982	1984	1982	1982

engineer were modified at the time of plant integration to update the cost and power related to cooling requirements. These changes used the Base Case 1, and then ratioed the wet cooling tower cost and the power for cooling tower fan drive to the rate at which steam was condensed.

RESULTS

The parametric variations studied are presented in Table 2.2-1 with the resulting capital cost distributions in Table 2.2-2. The configurations were four gas turbines and one bottoming steam turbine except for Case 2 with a single 1800 RPM gas turbine, Case 11 with two gas turbines, and Case 12 with eight gas turbines. The base conditions were 2200 F (1478 K) and a pressure ratio of 12 for the gas turbine, and 1250 psig (8.7 MN/m^2), 950 F (783 K) for the steam turbine condensing at 1.5 in. Hga (5.1 kN/m^2).

Rationale for Point Variations

The base case represents complete integration of gas turbine, steam turbine, and coal gasification at conditions appropriate for realization in the near term. The level of water consumption imposed by coal gasification justifies some additional use in wet cooling towers for the most advantageous steam turbine conditions.

Table 2.2-3 presents the base case parameters along with the range of variations that were explored. Tables 2.2-4 and 2.2-5 indicate the groupings of cases that were used to explore the significant variables of the base case power plant.

These groupings provide for determination of sensitivities of plant efficiency and cost of power generation to the dominant variables that may be arbitrarily set. The higher firing temperatures such as 2600 F (1700 K) are deemed to be future developments as are the use of ceramic stationary parts in gas turbines. In order to effectively produce steam for a reheat steam turbine, the gas inlet temperature to the HRSG should be 1300 F (978 K) or higher. Fuel firing in the HRSG can be used as a supplemental energy source to raise the exhaust gas temperature in order to obtain effective HRSG operation. Such firing was limited by either a 1400 F (1033 K) maximum temperature or oxygen depletion to the equivalent of 25 percent excess air, whichever proved most stringent.

An intercooled gas turbine compressor requires a two-section compressor. Only the machines with a pressure ratio of 20 are naturally of this configuration. At lower pressure ratios the added expense and complexity of a two-spool compressor-turbine configuration cannot be justified. A single case at a pressure ratio of 20 with 2600 F((1700 K) firing temperature was evaluated.

Table 2.2-5 shows that most gas turbine variations were at a pressure ratio of 12. In addition a span of pressures and other conditions were explored at the 2600 F (1700 K) firing

Table 2.2-2 (Page 1 of 4)

CAPITAL COST DISTRIBUTIONS FOR OPEN-CYCLE GAS TURBINE COMBINED CYCLE—AIR COOLED

	CASE NO.	1	2	3	4	5	6	7	8	9	10
MAJOR COMPONENTS											
PRIME CYCLE											
GAS TURB-COMP-COMB-GEN	MM$	28.4	27.3	28.7	28.6	26.0	26.0	26.0	25.2	29.1	29.2
BOTTOMING CYCLE											
HEAT RECOVERY STEAM GEN (HRSG)	MM$	16.0	16.1	16.0	16.0	11.6	11.6	11.6	11.6	10.4	11.6
STEAM TURB-GEN	MM$	7.0	7.0	7.1	7.1	7.2	7.2	7.2	5.9	5.9	5.9
PRIMARY HEAT INPUT AND FUEL SYSTEM											
GASIFIER (INCLUDING BOOST STEAM TURB-COMP	MM$	71.0	71.0	84.0	77.0	0.	0.	0.	0.	0.	0.
SUB-TOTAL OF MAJOR COMPONENTS	MM$	122.4	121.4	135.8	128.7	44.8	44.8	44.8	42.7	45.4	46.7
BALANCE OF PLANT											
COOLING TOWER	MM$	1.3	1.3	1.2	1.2	1.7	1.7	1.7	1.5	1.5	1.5
ALL OTHER	MM$	30.8	30.8	34.7	32.1	32.0	32.0	37.0	21.6	24.6	24.6
SITE LABOR	MM$	11.7	11.7	13.3	12.3	12.3	12.3	12.3	8.5	8.8	8.8
SUB-TOTAL OF BALANCE OF PLANT	MM$	43.8	43.8	49.2	45.6	46.0	46.0	46.0	31.6	34.9	34.9
CONTINGENCY	MM$	33.2	33.[illegible]	37.0	34.9	18.2	18.2	18.2	14.9	16.1	16.3
ESCALATION COSTS	MM$	30.2	30.0	33.6	31.6	12.2	12.2	12.2	10.0	14.6	14.8
INTEREST DURING CONSTRUCTION	MM$	29.6	29.7	33.2	31.3	10.0	10.0	10.0	8.2	14.4	14.6
TOTAL CAPITAL COST	MM$	259.4	258.0	288.9	272.0	131.2	131.2	131.2	107.4	125.4	127.3
MAJOR COMPONENTS COST	$/KWE	212.5	210.9	226.7	217.4	87.3	86.2	86.6	89.8	99.5	101.3
BALANCE OF PLANT	$/KWE	76.1	76.1	82.2	77.0	89.6	88.5	89.0	66.5	76.4	75.8
CONTINGENCY	$/KWE	57.7	57.4	61.8	58.9	35.4	34.9	35.1	31.3	35.2	35.4
ESCALATION COSTS	$/KWE	52.4	52.1	56.1	53.4	23.7	23.4	23.6	21.0	31.9	32.1
TOTAL CAPITAL COST	$/KWE	450.6	448.1	482.2	459.6	255.5	252.3	253.7	225.8	274.6	276.5

Table 2.2-2 (Page 2 of 4)

CAPITAL COST DISTRIBUTIONS FOR OPEN-CYCLE GAS TURBINE COMBINED CYCLE—AIR COOLED

	CASE NO.	11	12	13	14	15	16	17	18	19	20
MAJOR COMPONENTS											
PRIME CYCLE											
GAS TURB-COMP-COMB-GEN	MM$	14.2	56.7	26.1	33.6	38.9	40.6	34.8	34.8	30.4	49.8
BOTTOMING CYCLE											
HEAT RECOVERY STEAM GEN (HRSG)	MM$	8.0	32.0	16.4	16.0	16.0	16.0	18.0	15.2	16.4	16.0
STEAM TURB-GEN	MM$	5.6	13.1	6.4	8.6	9.1	8.3	10.0	10.1	6.3	5.2
PRIMARY HEAT INPUT AND FUEL SYSTEM											
GASIFIER (INCLUDING BOOST STEAM TURB-COMP	MM$	41.0	128.0	62.0	80.0	89.0	74.0	95.0	95.0	60.0	84.0
SUB-TOTAL OF MAJOR COMPONENTS	MM$	68.8	229.8	110.9	138.2	153.0	138.9	157.8	155.1	113.1	155.0
BALANCE OF PLANT											
COOLING TOWER	MM$	0.6	2.5	1.1	1.5	1.7	1.4	2.0	2.1	1.0	0.7
ALL OTHER	MM$	15.8	61.0	30.9	30.8	30.8	30.8	30.7	30.7	30.9	30.9
SITE LABOR	MM$	5.9	23.0	11.7	11.7	11.7	11.7	11.7	11.7	11.7	11.7
SUB-TOTAL OF BALANCE OF PLANT	MM$	22.3	86.5	43.7	44.0	44.2	43.9	44.4	44.5	43.6	43.3
CONTINGENCY	MM$	18.2	63.3	30.9	36.4	39.4	36.6	40.4	39.9	31.3	39.7
ESCALATION COSTS	MM$	16.5	73.7	28.0	33.1	35.8	33.2	36.7	36.2	28.4	36.0
INTEREST DURING CONSTRUCTION	MM$	16.4	80.0	27.7	32.7	35.4	32.8	36.3	35.8	28.1	35.6
TOTAL CAPITAL COST	MM$	142.2	533.3	241.2	284.5	307.8	285.4	315.6	311.6	244.6	309.5
MAJOR COMPONENTS COST	$/KWE	241.0	198.9	230.5	204.3	198.3	202.1	209.8	201.6	221.5	229.0
BALANCE OF PLANT	$/KWE	78.3	74.9	90.8	65.1	57.3	63.9	59.0	57.8	85.4	64.0
CONTINGENCY	$/KWE	63.9	54.8	64.3	53.9	51.1	53.2	53.8	51.9	61.4	58.6
ESCALATION COSTS	$/KWE	58.0	63.8	58.3	48.9	46.4	48.3	48.8	47.1	55.7	53.2
INTEREST DURING CONSTRUCTION	$/KWE	57.3	69.3	57.7	48.3	45.9	47.8	48.3	46.6	55.1	52.6
TOTAL CAPITAL COST	$/KWE	498.4	461.6	501.6	420.4	399.0	415.3	419.6	404.9	479.2	457.5

CAPITAL COST DISTRIBUTIONS FOR OPEN-CYCLE GAS TURBINE COMBINED CYCLE—AIR COOLED

	CASE NO.	21	22	23	24	25	26	27	30	31	32
MAJOR COMPONENTS											
PRIME CYCLE											
GAS TURB-COMP-COMB-GEN	MM$	45.3	28.4	28.3	28.4	28.4	28.4	28.4	28.4	28.4	28.4
BOTTOMING CYCLE											
HEAT RECOVERY STEAM GEN (HRSG)	MM$	16.0	12.4	11.2	16.0	16.4	16.0	16.0	14.8	14.8	17.6
STEAM TURB-GEN	MM$	6.7	7.0	7.0	6.9	7.1	7.0	7.9	6.7	6.7	9.3
PRIMARY HEAT INPUT AND FUEL SYSTEM											
GASIFIER (INCLUDING BOOST STEAM TURB-COMP	MM$	72.0	71.0	71.0	71.0	71.0	71.0	71.0	71.0	71.0	82.0
SUB-TOTAL OF MAJOR COMPONENTS	MM$	140.0	118.8	117.5	122.3	122.9	122.4	123.3	120.9	120.9	137.3
BALANCE OF PLANT											
COOLING TOWER	MM$	1.1	1.2	1.3	1.3	1.2	1.3	1.3	1.3	1.3	1.9
ALL OTHER	MM$	30.9	30.8	30.8	30.8	30.8	30.8	30.8	30.8	30.8	30.7
SITE LABOR	MM$	11.7	11.7	11.7	11.7	11.7	11.7	11.7	11.7	11.7	11.7
SUB-TOTAL OF BALANCE OF PLANT	MM$	43.7	43.8	43.8	43.8	43.8	43.8	43.8	43.8	43.8	44.4
CONTINGENCY	MM$	36.7	32.5	32.3	33.2	33.3	33.2	33.4	32.9	32.9	36.3
ESCALATION COSTS	MM$	33.3	29.5	29.3	30.1	30.2	30.2	30.3	29.9	29.9	33.0
INTEREST DURING CONSTRUCTION	MM$	33.0	29.2	29.0	29.8	29.9	29.8	30.0	29.6	29.6	32.6
TOTAL CAPITAL COST	MM$	286.7	253.9	251.8	259.3	260.1	259.4	260.8	257.1	257.1	283.5
MAJOR COMPONENTS COST	$/KWE	235.6	205.1	205.3	212.9	212.9	214.2	212.9	213.0	213.1	212.0
BALANCE OF PLANT	$/KWE	73.5	75.6	76.6	76.3	75.8	76.7	75.7	77.3	77.3	68.5
CONTINGENCY	$/KWE	61.8	56.1	56.4	57.8	57.7	58.2	57.7	58.1	58.1	56.1
ESCALATION COSTS	$/KWE	56.1	50.9	51.2	52.5	52.4	52.8	52.4	52.7	52.7	50.9
INTEREST DURING CONSTRUCTION	$/KWE	55.5	50.4	50.6	51.9	51.8	52.2	51.8	52.1	52.1	50.4
TOTAL CAPITAL COST	$/KWE	482.5	438.2	440.0	451.4	450.7	454.0	450.5	453.1	453.3	437.9

Table 2.2-2 (Page 4 of 4)

CAPITAL COST DISTRIBUTIONS FOR OPEN-CYCLE GAS TURBINE COMBINED CYCLE—AIR COOLED

	CASE NO.	33	34	35	36	37
MAJOR COMPONENTS						
PRIME CYCLE						
GAS TURB-COMP-COMB-GEN	MM$	28.4	28.4	38.9	41.0	28.4
BOTTOMING CYCLE						
HEAT RECOVERY STEAM GEN (HRSG)	MM$	16.0	16.8	15.6	16.0	16.0
STEAM TURB-GEN	MM$	6.8	10.0	9.5	8.4	7.0
PRIMARY HEAT INPUT AND FUEL SYSTEM						
GASIFIER (INCLUDING BOOST STEAM TURB-COMP	MM$	71.0	82.0	89.0	83.0	71.0
SUB-TOTAL OF MAJOR COMPONENTS	MM$	122.2	137.2	153.0	148.4	122.4
BALANCE OF PLANT						
COOLING TOWER	MM$	1.4	1.8	1.7	1.0	5.0
ALL OTHER	MM$	32.2	30.7	30.8	30.9	32.3
SITE LABOR	MM$	12.5	11.7	11.7	11.7	12.9
SUB-TOTAL OF BALANCE OF PLANT	MM$	46.1	44.3	44.2	43.6	50.2
CONTINGENCY	MM$	33.6	36.3	39.4	38.4	34.5
ESCALATION COSTS	MM$	30.5	32.9	35.8	34.8	31.3
INTEREST DURING CONSTRUCTION	MM$	30.2	32.6	35.4	34.5	31.0
TOTAL CAPITAL COST	MM$	262.6	283.2	307.7	299.7	269.3
MAJOR COMPONENTS COST	$/KWE	215.7	207.3	196.4	207.3	212.7
BALANCE OF PLANT	$/KWE	81.3	66.9	56.7	60.9	87.2
CONTINGENCY	$/KWE	59.4	54.8	50.6	53.6	60.0
ESCALATION COSTS	$/KWE	53.9	49.8	45.9	48.7	54.4
TOTAL CAPITAL COST	$/KWE	463.6	428.0	395.1	418.7	468.1

Table 2.2-3

AIR-COOLED COMBINED CYCLE GAS TURBINE EVALUATION

Parameter	Base Case	Variations
Power output (MW)	576	285, 1155
Application	Base Load	—
Number of coals	1	2
Coal conversion process	LBtu coal gas	Liquified SRC and COED, HBtu and IBtu gas
First-stage total temperature (°F)	2200	2000, 2400, 2600
Compressor pressure ratio	12	8, 16, 20
HRSG,* ΔP (in. H_2O)	20	10, 30
Steam turbine conditions:		
Throttle (psig)	1250	1000, 1450, 1500, 1800
Throttle (°F)	950	900, 1000
Reheat (°F)	—	950
Condenser (in. Hga)	1.5	1.9, 3.45

*HRSG = heat recovery steam generator; ΔP includes all exhaust ducting

Table 2.2-4

VARIABLES FOR AIR-COOLED COMBINED CYCLES

Variable	Case Numbers
Gas turbine size	1, 2
Fuel and coal source	LBtu 1, 3, 4 IBtu 5, 6, 7 HBtu 8 SRC 9 COED 10
Plant size	1, 11, 12
Gas turbine conditions	see table 2.2-5
Gas path pressure drop	1, 22, 23
Steam turbine conditions:	
Temperature	1, 24, 25, 35
Pressure	1, 26, 27, 35
Reheat	18, 34
HRSG fired	32, 34
HRSG pinch point	1, 30, 31
Dry cooling tower	33, 37

Table 2.2-5

AIR-COOLED GAS TURBINE CASES FOR VARIATION OF PRESSURE RATIO AND TEMPERATURE

First-stage Total Temperature (°F)	Compressor Pressure Ratio			
	8	12	16	20
2000	—	(13)*	—	—
2200	—	(1)	(19)	—
2400	—	(14)	—	—
2600	(17)	(15)	(16)	(21)
2600, Ceramics	—	—	(36)	—
2600, Intercooled compressor	—	—	—	(20)

*Numerals in ( ) are case numbers.

temperature. This latter was construed to be the probable economic limit for air-cooled gas turbines as compared with the potential offered by water-cooled gas turbined of advanced design.

Base Case Results

The summary of all performance parameters including size and cost of major components and environmental intrusions for the base case are presented in Table 2.2-6. It is apparent that the gasification system dominates the major component cost. The resulting plant capital cost was low as compared with other plants considered in the entire study. The cost of power generation, 25.5 mills/kWh, was one of the lowest found. The total water consumed in gallons per kilowatt hour was approximately 50 percent greater than that required by steam plants. The emissions were of the order of half the pounds per kilowatt hour realized in a steam power plant with an atmospheric fluidized bed (AFB) combustion system.

Other Results

Table 2.2-7 presents the makeup of power generation by gas turbines and by the bottoming steam turbine and the power consumed by auxiliaries to form the net station output on a case by case basis.

Table 2.2-6

SUMMARY SHEET
OPEN-CYCLE GAS TURBINE COMBINED CYCLE— AIR-COOLED BASE CASE

CYCLE PARAMETER

Net Power Output (MWe)	576
Coal Type and Conversion Process	Illinois No. 6 LBtu
Prime Cycle	
Turbine inlet temperature (°F)	2200
Compressor presure ratio	12
Gas path Δp (in. H_2O)	20
Bottoming Cycle	
Turbine inlet temperature (°F)	950
Turbine inlet pressure (psi)	1250
Condenser (in. Hga)	1.5
Heat Exchanger	
ΔT gas to super heater (°F)	187
ΔT boiler pinch point (°F)	30
Supplemental firing	0
Heat Rejection	Wet cooling tower

PERFORMANCE AND COST

Thermodynamic efficiency (percent)	--
Powerplant efficiency (percent)	35.6
Overall energy efficiency (percent)	35.6
Plant capital cost ($ x 10^6)	226
Plant capital cost ($/kWe)	450
Cost of electricity (mills/kWh)	25.5

NATURAL RESOURCES

Coal (lb/kWh)	0.88
Water (gal/kWh)	
Total	0.60
Cooling	0.40
Processing	0.08
Makeup	0.12
NO_x suppression	0
Stack gas cleanup	0
Land (acres/100 MWe)	8.95

ENVIRONMENTAL INTRUSION

	Lb/10^6-Btu Input	Btu/kWh	Lb/kWh Output
Gas Turbines			
SO_2	0.05		0.47×10^{-3}
NO_x	0.14		1.33×10^{-3}
HC	0		0
CO	0		0
Particulates	0		0
Gasifier System			
SO_2	0.20		1.89×10^{-3}
NO_x	0		0
Heat to water		1410	
Heat, total rejected		6172	

	Lb/kWh	Lb/Day
Wastes		
Elemental sulfur	33.4×10^{-3}	0.466×10^6
Ash (wet)	99.1×10^{-3}	1.385×10^6

MAJOR COMPONENT CHARACTERISTICS

Major Component	Unit or Module: Size (ft) (W x L (or D) x H)	Unit or Module: Weight (lb) (x 10^6)	Unit or Module: Cost ($ x 10^6)	Units Required	Total Cost ($ x 10^6)	$/kW Output
Gas turbine-compressor-combustor-generator	14.5 x 63 x 17	0.62	7.1	4	28.4	48.8
Heat recovery steam generator	40 x 37 x 63	0.587	4.0	4	16.0	27.5
Steam turbine-generator	89 x 12 x 14	0.97	7.0	1	7.0	12.0
Gasifier system (including booster and steam turbine-compressor				--	71.0	122.0

Table 2.2-7

POWER OUTPUT AND AUXILIARY POWER DEMAND FOR BASE CASE AND PARAMETRIC VARIATIONS: OPEN-CYCLE GAS TURBINE COMBINED CYCLE-AIR COOLED

CASE NO.		1	2	3	4	5	6	7	8	9	10
PRIME CYCLE POWER OUTPUT	MW	449.2	449.2	473.6	465.2	368.0	373.2	370.4	346.8	332.4	334.4
BOTTOMING CYCLE POWER OUTPUT	MW	149.9	149.9	148.4	148.0	163.0	164.4	164.2	145.8	141.0	142.8
FURNACE POWER OUTPUT	MW	0.	0.	0.	0.	0.	0.	0.	0.	0.	0.
BALANCE OF PLANT AUX. POWER REQ'D.	MW	14.4	14.4	14.2	14.2	14.6	14.6	14.6	14.4	14.3	14.3
FURNACE AUX. POWER REQ'D.	MW	6.0	6.0	5.5	4.1	0.	0.	0.	0.	0.	0.
TRANSFORMER LOSSES	MW	3.0	3.0	[illegible].1	3.1	2.7	2.7	2.7	2.5	2.4	2.4
NET STATION OUTPUT	MW	575.7	575.7	599.1	591.8	513.7	520.3	517.2	475.8	456.7	460.5

CASE NO.		11	12	13	14	15	16	17	18	19	20
PRIME CYCLE POWER OUTPUT	MW	224.6	898.4	380.0	519.2	587.2	539.6	537.6	537.6	413.2	598.8
BOTTOMING CYCLE POWER OUTPUT	MW	73.3	301.0	122.6	182.5	211.1	172.7	242.1	260.3	119.2	102.1
FURNACE POWER OUTPUT	MW	0.	0.	0.	0.	0.	0.	0.	0.	0.	0.
BALANCE OF PLANT AUX. POWER REQ'D.	MW	8.1	26.1	14.1	14.7	15.0	14.6	15.3	16.0	14.0	13.6
FURNACE AUX. POWER REQ'D.	MW	3.0	12.0	5.1	6.9	7.8	6.8	8.3	8.3	5.2	7.2
TRANSFORMER LOSSES	MW	1.5	6.0	2.5	3.5	4.0	3.6	3.9	4.0	2.7	3.5
NET STATION OUTPUT	MW	285.3	1155.3	480.9	676.6	771.4	687.3	752.2	769.7	510.5	676.6

CASE NO.		21	22	23	24	25	26	27	30	31	32
PRIME CYCLE POWER OUTPUT	MW	486.0	455.6	443.2	449.2	449.2	449.2	449.2	449.2	449.2	449.2
BOTTOMING CYCLE POWER OUTPUT	MW	131.3	147.1	152.4	148.6	151.2	145.4	153.2	141.5	141.4	224.0
FURNACE POWER OUTPUT	MW	0.	0.	0.	0.	0.	0.	0.	0.	0.	0.
BALANCE OF PLANT AUX. POWER REQ'D.	MW	14.1	14.3	14.4	14.4	14.3	14.4	14.4	14.4	14.4	15.3
FURNACE AUX. POWER REQ'D.	MW	5.9	6.0	6.0	6.0	6.0	6.0	6.0	6.0	6.0	7.1
TRANSFORMER LOSSES	MW	3.1	3.0	3.0	3.0	3.0	3.0	3.0	3.0	3.0	3.4
NET STATION OUTPUT	MW	594.2	579.4	572.2	574.4	577.1	571.3	579.0	567.3	567.2	647.5

CASE NO.		33	34	35	36	37
PRIME CYCLE POWER OUTPUT	MW	449.2	449.2	587.2	562.0	449.2
BOTTOMING CYCLE POWER OUTPUT	MW	140.7	238.7	218.5	178.4	149.9
FURNACE POWER OUTPUT	MW	0.	0.	0.	0.	0.
BALANCE OF PLANT AUX. POWER REQ'D.	MW	14.5	15.6	15.0	14.0	14.8
FURNACE AUX. POWER REQ'D.	MW	6.0	7.1	7.8	7.0	6.0
TRANSFORMER LOSSES	MW	2.9	3.4	4.0	3.7	3.0
NET STATION OUTPUT	MW	566.4	661.8	778.9	715.7	575.3

DISCUSSION OF RESULTS

Gas Turbine Size and Plant Size

No economic advantage was found for the 1800 RPM single gas turbine as compared with four 3600 RPM gas turbines in Cases 1 and 2. Since the four units should experience a higher plant availability, there is little incentive for using an 1800 RPM gas turbine at 448 MW.

Plant size effects are compared at base case conditions in Table 2.2-8. The double size plant of Case 12 incurs a greater interest during construction and escalation as a result of the added year of construction, with the cost of power generation being comparable to the base case. The half size plant has a modest economic penalty that should not foreclose consideration of this alternative. This type of power plant is highly modularized in its gasifiers, gas turbines, and HRSGs. As a consequence the sensitivity to plant size is low.

Table 2.2-8

PLANT SIZE EFFECTS FOR AIR-COOLED GAS TURBINE CASES

Plant Size	Output MW (Case)	$/kW	Cost mills/kWh	Years to Build
Half	285 (11)	498	27.7	3
Base	576 (1)	450	25.5	3
Double	1155 (12)	461	25.4	4

Fuel and Configuration Influence

The influence of the plant configuration and the fuel for the gas turbine at base plant conditions is summarized in Table 2.2-9. The least generation cost, 24.8 mills/kWh, was realized for firing any of the intermediate-Btu (IBtu) gases or for liquid solvent refined coal. Both of these plants receive their fuels from off-site. The IBtu gas was a unique fuel insofar as neither water nor steam was required for suppression of NO_X during combustion. The SRC represented the opposite extreme in that the fuel-bound nitrogen produced more NO_X than permitted by the current emission standards.

The low-Btu gasification showed comparable costs for Illinois No. 6 coal and for Montana Sub-bituminous coal. The North Dakota Lignite of Case 3 showed an increased generation cost of 1.1 mills/kWh.

The least electricity cost, 22.9 mills/kWh, was 10 percent less than the base case, but the gas turbine firing temperature would be 2600 F in place of 2200 F.

Table 2.2-9

CONFIGURATION AND COAL INFLUENCE ON ELECTRICITY COST FOR AIR-COOLED GAS TURBINE CASES

Configuration	Cost-Various Coal Sources (mills/kWh)			
	Ill.	Mont.	N.D.	Other
Base Case:				
Low-Btu gasification	25.5 (1)*	25.4 (4)	26.6 (3)	—
Intermediate-Btu gas produced off-site	24.8 (5)	24.8 (6)	24.8 (7)	—
High-Btu gas—off-site	30.4 (8)	—	—	—
Liquid COED—off-site	—	—	—	30.2 (10)
Liquid SRC—off-site	—	—	—	24.8 (9)
Least Cost Case:				
Low-Btu gasification**	22.9 (35)**	—	—	—

*Numerals in ( ) are case numbers.
**2600 F, 12 pressure ratio gas turbine; 1450 psig, 1000 F steam turbine.

Gas Turbine Temperature Effects

The generation costs for the group of cases at a pressure ratio of 12, from Table 2.2-5, are presented in Figure 2.2-2 as a function of gas turbine temperature. This figure illustrates the progressive reduction in generation cost as turbine temperature increases. The cost at 2000 F (1366 K) was below the least cost for steam plants.

The effect for shifting to a dry cooling tower was found to be an increment of 0.6 mills/kWh. The shift to a reheat steam turbine with firing of low-Btu gas to 1400 F (1033 K) at the HRSG provides a reduction of 0.6 mills/kWh.

Gas Turbine Pressure Ratio Effects

Figure 2.2-3 presents the remainder of the survey tabulated in Table 2.2-5 in the context of variation of compressor pressure ratio at a 2600 F (1700 K) gas turbine temperature. The

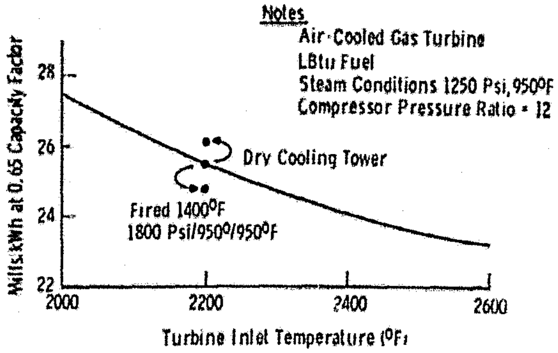

Figure 2.2-2. Turbine Temperature Effects

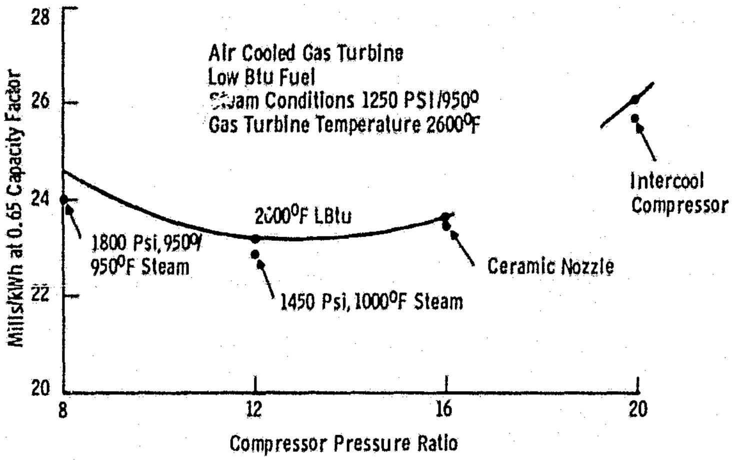

Figure 2.2-3. Gas Turbine Pressure Ratio Effects

minimum cost was found at a pressure ratio of 12. The intercooled compressor was advantageous at a pressure ratio of 20; it is not applicable to the single spool compressors at lower pressure ratios. The advantage of the ceramic nozzles and stationary parts was found, but it was a modest advantage. Alternative steam conditions that may be produced at 2600 F (1700 K) firing temperature such as 1450 psig, 1000 F (10.1 MN/m^2, 811 K) or 1800 psig, 950 F/950 F (12.5 MN/m^2, 783 K/783 K) are advantageous.

Dry Power Plants

The combination of the plant burning IBtu gas produced offsite with dry cooling towers would have no water consumption. Although this was not one of the parametric points, its performance can be estimated by combining the dry cooling tower cost increment to the IBtu case. The resulting estimate is 25.4 mills/kWh for the free-standing gas turbine—steam turbine—dry cooling tower plant.

OBSERVATIONS

The gas turbine-steam turbine power plant integrates very well with the requirements for a coal gasification plant to produce low-Btu gas for combustion in the gas turbine. The capital cost was modest at approximately $450/kW. The pressure ratio of 12 for best efficiency is already in use. The cost of electricity was less than other advanced coal conversion power plants of this study even for cases with moderate gas turbine conditions.

There are many routes to progressive cycle and economic improvement such as higher steam conditions, firing fuel in the HRSG, increased firing temperature, ceramic nozzles, and gasifiers with reduced steam requirement and with hot gas cleanup.

RECOMMENDED CASES

A low-Btu integrated power plant with gas turbine pressure ratio of 12 and firing temperature of 2400 F (1589 K) and with a reheat steam turbine of 1800 psig, 950 F/950 F (12.5 MN/m^2, 783 K/783 K) is recommended. This recommendation corresponds most closely to Case 14 except for the more advanced steam conditions. It may prove necessary to fire low-Btu gas in the HRSG to obtain better steam conditions.

A second recommendation is a free-standing combined plant burning liquid solvent refined coal at 2400 F (1589 K) with a pressure ratio of 12 and steam conditions that do not require fuel firing in the HRSG.

Generation cost of 23.4 mills/kWh was estimated for these recommended cases.

2.3 OPEN-CYCLE GAS TURBINE COMBINED CYCLE—WATER COOLED

DESCRIPTION OF CYCLE

The schematic for this power plant, Figure 2.3-1, is very similar to the integrated air-cooled gas turbine plant with low-Btu gasifier. The similarities should be noted first so that emphasis can thereafter focus on the differences. The deployment of the gas turbines, booster compressor, gasifier, heat recovery steam generator (HRSG), bottoming steam turbine, and wet cooling towers are identical to that shown in Figure 2.2-1. The feedwater heating train and the gas turbine cooling system are very different and represent a high level of integration into the total system.

The gas turbine schematic shows blocking air flow from the compressor to the gas turbine. This is a very small flow that continually purges heated gas out of rotor spaces and blocks the intrusion of hot gases. The stationary parts of the gas turbine hot gas path are cooled by pressurized water in a circuit that is integral with the steam turbine feedwater system. These parts are the combustor transition piece and the nozzles for each turbine stage. The heated water is throttled into a flash tank that sends steam to the gasifier. Thereafter the flash tank water is throttled into the deaerating feedwater heater, thus reducing the steam extracted from the steam turbine for that purpose.

The system for water cooling the rotating buckets to bring their surface temperature to about 900 F (755 K) must provide protection to the steam system from contamination from water that has been exposed to gas turbine combustion gases. High purity water at about 100 F (311 K) is pumped through passages beneath the bucket skin. A portion of the coolant water is converted to steam as the water passes through the bucket. The coolant water-steam mixture is expanded through a nozzle at the outer tip of the moving bucket to produce a jet that assists in driving the bucket. The steam formed merges with the gas stream passing through the turbine along with a small fraction of the coolant water. Most of the coolant water is carried by centrifugal force to the outer diameter of the stationary casing. There it is collected for recycling. This water has been exposed to turbine gases and may be contaminated. The recovered heated water is not merged with other water but is cooled in a feedwater heater for the steam cycle. The recirculating water and makeup water are continuously polished to limit the level of contamination.

When ceramic stationary parts are considered for this gas turbine, the water-cooled bucket system is unchanged. Only the water coolant for the transition piece and the nozzles is eliminated.

The basic configuration used three gas turbines with 700 lb/s airflow each (317.5 kg/s) and a single steam turbine with

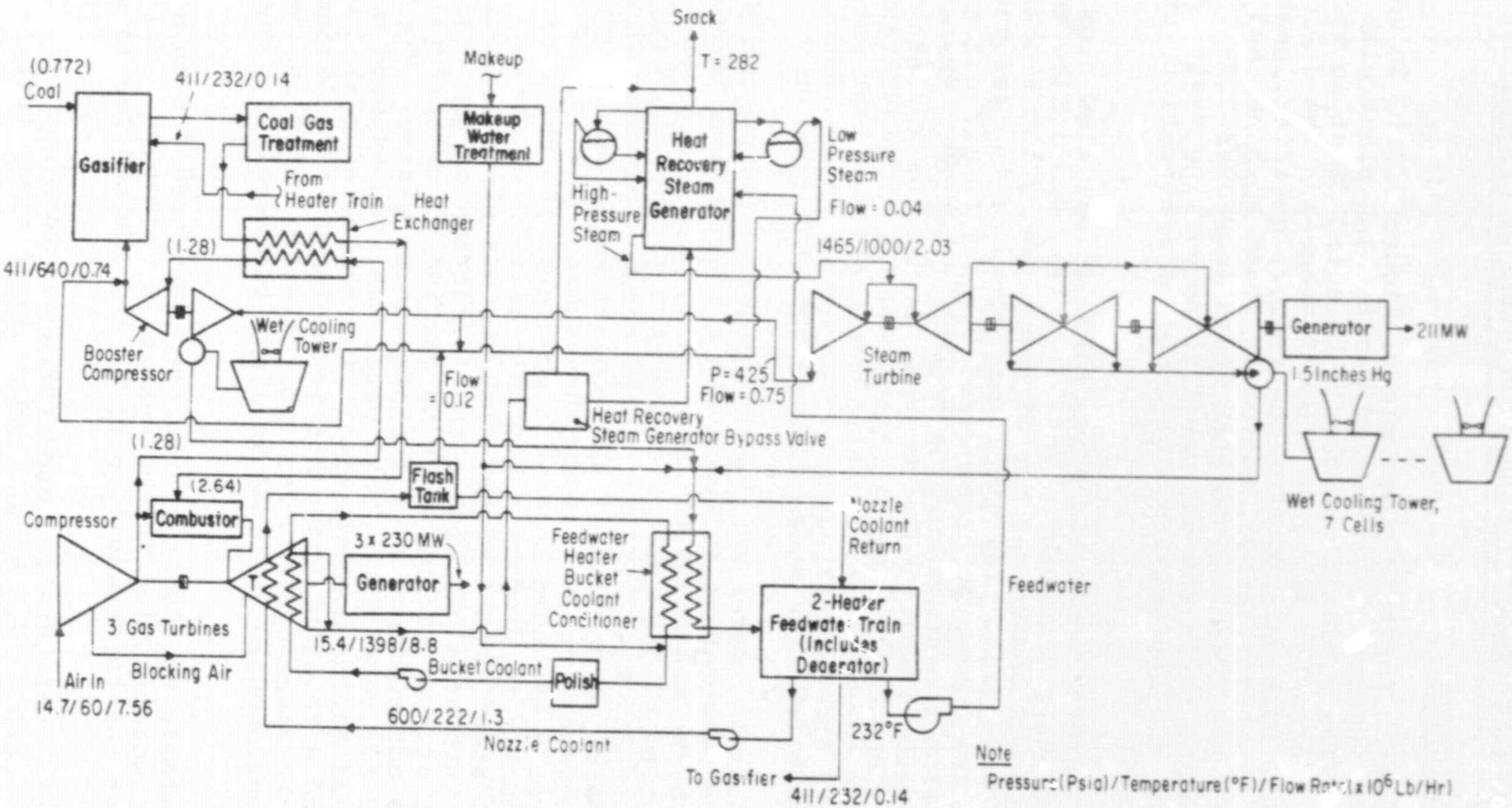

Figure 2.3-1. Open-Cycle Gas Turbine Combined Cycle—Water Cooled

1450 psig, 1000 F (10.1 MN/m^2, 811 K) conditions for the base case. When a reheat steam turbine is used, the HRSG is constructed without the low pressure drum for making gasifier steam.

The level of plant integration was greatly reduced for those cases considering free-standing plants burning high-Btu fuel gas, intermediate-Btu fuel gas, and liquified coal. The NO_X suppression steam required at the gas turbine combustors was drawn from the flash tank in the gas turbine cooling circuit. As a result of the elimination of the steam requirement for the booster compressor drive and for the gasifier, the bottoming steam turbine was conventional in the sense that great quantities of steam were not extracted from it.

ANALYTICAL PROCEDURES AND ASSUMPTIONS

The component and gas turbine cycle assumptions are the same for the water-cooled gas turbine as those for the air-cooled gas turbines of sections 2.1 and 2.2. One significant difference is the strict accounting for heat transfer from the hot gas to the surrounding passages at all stages of combustion and gas expansion. The firing temperature at the combustor is appreciably higher than the total temperature at the first turbine stage to allow for heat loss to combustor, transition piece, and first-stage nozzle coolant. The detailed water-cooling effect is evuluated along the gas path in the turbine so that the flow is treated as a real nonadiabatic expansion. The turbine exhaust conditions account for the added water vapor from unrecovered bucket coolant as well as the small air coolant addition.

The heat loss due to water cooling was calculated for each element in the cooled gas path; nozzle blading and wall interstage passage, bucket, bucket shroud, wall, platform, transition piece, and turbine exit diffuser. This heat loss was based on gas temperature, metal surface temperature, and stage geometry and loading parameters. Changes in stage efficiency due to nonadiabatic bucket flow and increased tip leakage due to the low temperature thermal boundary layer of the leakage at the wall were also considered.

The work required to bring the bucket coolant up to bucket tip speed and the work recovered when this coolant was expanded through the tip jets was calculated and considered as part of the turbine output. Blockage air and unrecovered bucket coolant were treated as mixed between stages to determine the gas properties entering the subsequent turbine stage and for the turbine exhaust flow.

The basic gas turbine has a compressor inlet airflow of 700 lb/s (317.5 kg/s) and operates at 3600 RPM. The expansion turbine has three stages for pressure ratios of 16 and below, and four stages at a pressure ratio of 20.

DESIGN AND COST BASIS

The design and cost basis described for the air-cooled gas turbine combined plants are exactly those followed for the water-cooled combined cycle gas turbine plants. The water-cooled gas turbine cost was evaluated in detail for the base case. Thereafter adjustments were made for the alternative gas turbine conditions. The very low temperatures for hot gas path surfaces realized with this mode of cooling results in a high retention of material strength irrespective of firing temperature. Accordingly the maximum temperatures represent combustion limits rather than hot rotating part limits.

RESULTS

The parametric variations that were studied are presented in Table 2.3-1 with the resulting capital cost distributions in Table 2.3-2. The basic configuration had three gas turbines with alternatives of two and four being considered in Cases 7 and 8. Base case gas turbine conditions were 2800 F (1811 K) and 16 pressure ratio; for the single bottoming steam turbine conditions were 1450 psig, 1000 F (10.1 MN/m^2, 811 K), condensing at 1.5 in. Hga (5.1 kN/m^2).

Rationale for Point Variations

The water-cooled gas turbines were considered at high output powers appropriate to their greater inlet airflow and their high firing temperature. Pressure ratios were selected to maximize the gas turbine output power. The steam bottoming cycles used higher pressures and temperatures than formerly in order to exploit fully the increased exhaust temperature from the water-cooled g turbine. More examples of reheat steam turbine cycles were explored; firing supplemental fuel in the HRSG was not needed to sustain reheat operation because of temperatures near 1400 F (1033 K) entering the HRSG.

Table 2.3-3 presents the base case parameters and the range of variations that were explored. The limited scope for each variable explored makes the upper half of Table 2.3-1 sufficient to indicate the interplay among the 28 cases.

The several groupings provide for exploration of sensitivities of plant efficiency and cost of electricity to the significant plant parameters that may be arbitrarily set.

Base Case Results

Table 2.3-4 summarizes the base case results. As compared with Table 2.2-6 for the air-cooled base case, the performance results differ very little. The overall energy efficiency was marginally better as was the cost of electricity and the consumption of water. For the emissions the NO_x discharge was

Parameters	Case 1*	2
Power Output (MWe)	865	893
Coal and Conversion Process	Ill. #6 LBtu	N.D. LBtu
Prime Cycle		
Turbine inlet temperature (°F)	2800	
Compressor pressure ratio	16	
Gas path (Δp in. H_2O)	20	
Bottoming Cycle		
Turbine inlet temperature (°F)	1000	
Turbine inlet pressure (psi)	1450	
Condenser (in. Hga)	1.5	
Heat Exchanger		
ΔT gas to superheat (minimum)	50	
ΔT boiler pinch point (nominal)	30	
Supplemental firing	0	
Heat Rejection	WCT	
Actual Powerplant Output (MWe)	865	893
Thermodynamic Efficiency (percent)	0.	0.
Powerplant Efficiency (percent)	35.5	35.0
Overall Energy Efficiency (percent)	35.5	35.0
Coal Consumption (lb/kWh)	0.89	1.42
Plant Capital Cost ($ million)	378	423
Plant Capital Cost ($/kWe)	436	474
Cost of Electricity, Capacity Factor = 0.65		
Capital (mills/kWh)	13.8	15.0
Fuel (mills/kWh)	8.2	8.3
Maintenance and operating (mills/kWh)	3.2	3.3
Total (mills/kWh)	25.2	26.6
Sensitivity		
Capacity factor = 0.50 (total mills/kWh)	30.2	32.1
Capacity factor = 0.80 (total mills/kWh)	22.0	23.1
Capital Δ = 20 percent (Δ mills/kWh)	2.8	3.0
Fuel Δ = 20 percent (Δ mills/kWh)	1.6	1.7
Estimated Time for Construction (years)	4	4
Estimated Date of 1st Commercial Service (year)	1986	1986

*Base case -- Three gas turbines and one steam turbine, except cases 7 and 8.
**1000°F/1000°F reheat steam bottoming cycle.
† Ceramic transition piece and nozzle.

DCT =
HBtu =
Ill. =
LBtu =

Table 2.3-1

FOLDOUT FRAME 2

PARAMETRIC VARIATIONS FOR TASK I STUDY
OPEN-CYCLE GAS TURBINE COMBINED CYCLE—WATER COOLED

3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	20	23	24	25	26	27	28
886	762	728	735	580	1155	997	835	844	863	979	859	869	860	863	871	865	850	864	862	869	871
Mont LBtu	Ill. #6 HBtu	Ill. #6 SRC	Ill. #6 Coed	Ill. #6 LBtu																	→
					→	3000	2600	2800													→
						→	12	20	16	12	16†	16									→
											→	10	30	20							→
									**	**	** →	1000	→	850	1000					**	** →
														→	1800	1450			→	1800	2400
															→	1.9	3.45	1.5			→
																					→
																	→	50	70	30	→
																					→
															→	DCT	→	WCT			→
886	762	728	735	580	1155	997	835	844	863	979	859	869	860	863	871	865	850	864	862	869	871
0.	45.4	48.5	48.0	0.	0.	0.	0.	0.	0.	0.	0.	0.	0.	0.	0.	0.	0.	0.	0.	0.	0.
35.6	44.1	47.0	46.6	35.7	35.5	35.3	34.6	36.5	35.4	34.6	37.2	35.6	35.2	35.4	35.7	35.4	34.8	35.4	35.3	35.6	35.7
35.6	22.2	36.7	26.1	35.7	35.5	35.3	34.6	36.5	35.4	34.6	37.2	35.6	35.2	35.4	35.7	35.4	34.8	35.4	35.3	35.6	35.7
1.07	1.42	0.86	1.21	0.89	0.89	0.90	0.91	0.87	0.89	0.92	0.85	0.89	0.90	0.89	0.89	0.89	0.91	0.89	0.90	0.89	0.89
396	179	194	193	248	548	406	380	380	377	409	368	373	378	377	378	387	381	375	374	370	379
446	235	266	262	426	474	406	455	449	436	417	428	429	439	436	434	447	448	434	433	425	435
14.1	7.4	8.4	8.3	13.5	15.0	12.9	14.4	14.2	13.8	13.2	13.6	13.6	13.9	13.8	13.7	14.2	14.2	13.7	13.7	13.4	13.8
8.2	20.1	13.1	19.0	8.1	8.2	8.2	8.4	8.0	8.2	8.4	7.8	8.1	8.2	8.2	8.1	8.2	8.3	8.2	8.2	8.1	8.1
3.2	1.8	2.1	2.1	3.0	2.9	2.9	3.3	3.3	3.2	3.0	3.1	3.2	3.2	3.2	3.1	3.2	3.0	3.0	3.2	3.2	3.2
25.4	29.4	23.6	29.4	24.7	26.1	23.9	26.1	25.5	25.2	24.6	24.5	24.9	25.3	25.2	25.0	25.5	25.5	24.9	25.1	24.8	25.1
30.6	32.2	26.8	32.6	29.6	31.4	28.7	31.4	30.7	30.3	29.5	29.5	29.9	30.5	30.3	30.1	30.7	30.7	29.9	30.2	29.7	30.1
22.2	27.7	21.6	27.5	21.6	22.7	21.0	22.8	22.2	22.0	21.6	21.4	21.7	22.1	22.0	21.8	22.3	22.3	21.8	21.9	21.6	21.9
2.8	1.5	1.7	1.7	2.7	3.0	2.6	2.9	2.8	2.8	2.6	2.7	2.7	2.8	2.8	2.7	2.8	2.8	2.7	2.7	2.7	2.8
1.6	4.0	2.6	3.8	1.6	1.6	1.6	1.7	1.6	1.6	1.7	1.6	1.6	1.6	1.6	1.6	1.6	1.7	1.6	1.6	1.6	1.6
4	4	4	4	4	5	4	4	4	4	4	4	4	4	4	4	4	4	4	4	4	4
1986	1986	1988	1986	1986	1986	1987	1986	1986	1986	1986	1986	1986	1986	1986	1986	1986	1986	1986	1986	1986	1986

Dry cooling tower Mont = Montana
High Btu N.D. = North Dakota
Illinois SRC = Solvent refined coal
Low Btu WCT = Wet cooling tower

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Table 2.3-2 (Page 1 of 3)

CAPITAL COST DISTRIBUTIONS FOR OPEN-CYCLE GAS TURBINE COMBINED CYCLE—WATER COOLED

	CASE NO.	1	2	3	4	5	6	7	8	9	10
MAJOR COMPONENTS											
PRIME CYCLE											
GAS TURB-COMP-COMB-GEN	MM$	41.4	41.9	41.8	38.3	40.9	41.0	27.6	55.2	45.2	36.5
BOTTOMING CYCLE											
HEAT RECOVERY STEAM GEN (HRSG)	MM$	14.7	14.7	14.7	10.2	12.0	10.2	9.8	19.6	14.4	14.7
STEAM TURB-GEN	MM$	10.3	10.5	10.4	10.7	10.6	10.7	7.1	13.1	10.9	10.6
PRIMARY HEAT INPUT AND FUEL SYSTEM											
GASIFIER (INCLUDING BOOST STEAM TURB-COMP	MM$	93.0	111.0	100.0	0.	0.	0.	67.0	118.0	105.0	99.0
SUB-TOTAL OF MAJOR COMPONENTS	MM$	159.4	178.1	166.9	59.2	63.5	61.9	111.5	205.9	175.5	160.8
BALANCE OF PLANT											
COOLING TOWER	MM$	2.2	2.0	2.0	2.7	2.7	2.7	1.4	2.9	2.5	2.2
ALL OTHER	MM$	45.3	51.4	47.6	31.9	35.9	36.8	24.3	66.7	45.3	45.3
SITE LABOR	MM$	17.3	19.7	18.2	12.5	13.0	13.0	9.6	25.4	17.3	17.3
SUB-TOTAL OF BALANCE OF PLANT	MM$	64.8	73.1	67.8	47.1	51.6	52.5	35.4	95.0	65.1	64.9
CONTINGENCY	MM$	44.8	50.2	46.9	21.3	23.0	22.9	29.4	60.2	48.1	45.1
ESCALATION COSTS	MM$	52.2	58.5	54.6	24.8	26.8	26.6	34.2	86.3	56.0	52.6
INTEREST DURING CONSTRUCTION	MM$	56.7	63.5	59.4	26.9	29.1	28.9	37.2	100.4	60.9	57.1
TOTAL CAPITAL COST	MM$	378.0	423.4	395.7	179.3	194.1	192.9	247.6	547.8	405.5	380.5
MAJOR COMPONENTS COST	$/KWE	184.2	199.5	188.3	77.7	87.3	84.2	192.2	178.3	175.9	192.6
BALANCE OF PLANT	$/KWE	74.9	81.8	76.6	61.8	70.8	71.5	61.0	82.2	65.2	77.7
CONTINGENCY	$/KWE	51.8	56.3	53.0	27.9	31.6	31.1	50.6	52.1	48.2	54.1
ESCALATION COSTS	$/KWE	60.3	65.5	61.7	32.5	36.8	36.3	59.0	74.7	56.2	62.9
INTEREST DURING CONSTRUCTION	$/KWE	65.5	71.2	67.0	35.3	40.0	39.4	64.1	86.9	61.0	68.4
TOTAL CAPITAL COST	$/KWE	436.8	474.3	446.6	235.1	266.6	262.5	426.9	474.4	406.6	455.7

Table 2.3-2 (Page 2 of 3)

CAPITAL COST DISTRIBUTIONS FOR OPEN-CYCLE GAS TURBINE COMBINED CYCLE—WATER COOLED

	CASE NO.	11	12	13	14	15	16	17	20	23	24
MAJOR COMPONENTS											
PRIME CYCLE											
GAS TURB-COMP-COMB-GEN	MM$	47.3	41.4	39.5	41.4	41.5	41.3	41.4	41.4	41.4	41.4
BOTTOMING CYCLE											
HEAT RECOVERY STEAM GEN (HRSG)	MM$	12.0	13.2	13.2	13.8	11.7	14.7	14.1	14.7	14.7	14.7
STEAM TURB-GEN	MM$	9.0	9.7	10.6	9.8	10.3	10.4	10.2	10.3	10.3	9.0
PRIMARY HEAT INPUT AND FUEL SYSTEM											
GASIFIER (INCLUDING BOOST STEAM TURB-COMP	MM$	92.0	94.5	114.0	89.0	93.0	93.0	93.0	93.0	93.0	93.0
SUB-TOTAL OF MAJOR COMPONENTS	MM$	160.3	158.8	177.3	154.0	156.5	159.4	158.7	159.4	159.4	158.1
BALANCE OF PLANT											
COOLING TOWER	MM$	2.3	1.9	2.5	1.6	2.1	2.2	2.2	2.2	8.6	5.7
ALL OTHER	MM$	45.3	45.4	45.3	45.4	45.3	45.3	45.3	45.3	44.3	44.8
SITE LABOR	MM$	17.3	17.3	17.3	17.3	17.3	17.3	17.3	17.3	17.4	17.4
SUB-TOTAL OF BALANCE OF PLANT	MM$	64.9	64.6	65.1	64.4	64.8	64.8	64.9	64.8	70.3	67.9
CONTINGENCY	MM$	45.0	44.7	48.5	43.7	44.3	44.8	44.7	44.8	46.0	45.2
ESCALATION COSTS	MM$	52.4	52.0	56.4	50.8	51.5	52.2	52.1	52.2	53.5	52.6
INTEREST DURING CONSTRUCTION	MM$	57.0	56.5	61.3	55.2	56.0	56.7	56.6	56.7	58.1	57.2
TOTAL CAPITAL COST	MM$	379.6	376.7	408.6	368.1	373.1	378.0	376.9	378.0	387.3	381.1
MAJOR COMPONENTS COST	$/KWE	189.8	184.0	181.1	179.4	180.0	185.4	183.8	183.1	184.4	186.0
BALANCE OF PLANT	$/KWE	76.9	74.9	66.4	75.0	74.5	75.4	75.1	74.4	81.3	79.9
CONTINGENCY	$/KWE	53.3	51.8	49.5	50.9	50.9	52.2	51.8	51.5	53.1	53.2
ESCALATION COSTS	$/KWE	62.1	60.3	57.6	59.2	59.3	60.7	60.3	60.0	61.9	61.9
INTEREST DURING CONSTRUCTION	$/KWE	67.5	65.5	62.6	64.3	64.4	66.0	65.5	65.2	67.2	67.3
TOTAL CAPITAL COST	$/KWE	449.7	436.5	417.4	428.7	429.1	439.7	436.6	434.1	448.0	448.4

Table 2.3-2 (Page 3 of 3)

CAPITAL COST DISTRIBUTIONS FOR OPEN-CYCLE GAS TURBINE COMBINED CYCLE—WATER COOLED

	CASE NO.	25	26	27	28
MAJOR COMPONENTS					
PRIME CYCLE					
GAS TURB-COMP-COMB-GEN	MM$	41.4	41.4	41.4	41.4
BOTTOMING CYCLF					
HEAT RECOVERY STEAM GEN (HRSG)	MM$	13.2	12.6	10.6	15.9
STEAM TURB-GEN	MM$	10.1	9.9	9.7	10.0
PRIMARY HEAT INPUT AND FUEL SYSTEM					
GASIFIER (INCLUDING BOOST STEAM TURB-COMP	MM$	93.0	93.0	93.0	93.0
SUB-TOTAL OF MAJOR COMPONENTS	MM$	157.7	156.9	154.7	160.3
BALANCE OF PLANT					
COOLING TOWER	MM$	2.2	2.2	2.0	2.0
ALL OTHER	MM$	45.3	45.3	45.4	45.4
SITE LABOR	MM$	17.3	17.3	17.3	17.3
SUB-TOTAL OF BALANCE OF PLANT	MM$	64.8	64.9	64.6	64.7
CONTINGENCY	MM$	44.5	44.4	43.9	45.0
ESCALATION COSTS	MM$	51.8	51.6	51.1	52.4
INTEREST DURING CONSTRUCTION	MM$	56.3	56.1	55.5	56.9
TOTAL CAPITAL COST	MM$	375.2	373.9	369.8	379.3
MAJOR COMPONENTS COST	$/KWE	182.6	182.0	178.0	184.1
BALANCE OF PLANT	$/KWE	75.0	75.2	74.3	74.2
CONTINGENCY	$/KWE	51.5	51.5	50.5	51.7
ESCALATION COSTS	$/KWE	60.0	59.9	58.8	60.1
INTEREST DURING CONSTRUCTION	$/KWE	65.2	65.1	63.8	65.4
TOTAL CAPITAL COST	$/KWE	434.3	433.7	425.4	435.5

greater as a result of the higher firing temperature. The low capital cost and low cost of electricity are again evident.

Table 2.3-3

WATER-COOLED COMBINED CYCLE GAS TURBINE EVALUATION

	Base Case	Variations
Power output (MW)	865	580, 1155
Application	Base load	
Number of gas turbines	3	2, 4
Number of coals	1	2
Coal conversion process	LBtu coal gas	Liquefied SRC and COED, HBtu gas
First-stage total temperature °F	2800	2600, 3000
Compressor pressure ratio	16	12, 20
HRSG*, ΔP (in. H_2O)	20	10, 30
Steam turbine conditions:		
Throttle (psig)	1450	1800, 2400
Throttle (°F)	1000	950
Reheat (°F)	—	1000
Condenser (in. Hga)	1.5	1.9, 3.45

*HRSG = heat recovery steam generator; ΔP includes all exhaust ducting.

DISCUSSION OF RESULTS

The cost of electricity for most of the 28 cases studied was very low, in the range of 24.5 to 26.1 mills/kWh for all the low-Btu integrated power plants. The overall efficiency was highest for Case 14 with ceramic stationary parts at 37.2 percent and generally ranged within 1 percent of the base case value of 35.5 percent. The thermodynamic efficiency of the free-standing combined cycle plants burning liquid fuels was 48.5 and 48.0 percent. With such narrow ranges of variations the clearest understanding of the several influences at play are gained by examining the sensitivity to changes from the base cycle as presented in Table 2.3-5.

The influence of ceramic stationary parts on the water-cooled turbine were greater than those found for the air-cooled turbines. The heat transferred in the latter case warms air that re-enters the gas turbine hot gas path at the next downstream turbine stage.

Table 2.3-4

SUMMARY SHEET
OPEN-CYCLE GAS TURBINE COMBINED CYCLE—WATER COOLED BASE CASE

CYCLE PARAMETER

Net Power Output (MWe)	865
Coal Type and Conversion Process	Illinois No. 6 LBtu
Prime Cycle	
Turbine inlet temperature (°F)	2800
Compressor pressure ratio	16
Gas path Δp (in. H_2O)	20
Bottoming Cycle	
Turbine inlet temperature (°F)	1000
Turbine inlet pressure (psi)	1450
Condenser (in. Hga)	1.5
Heat Exchanger	
ΔT gas to super heat (°F)	398
ΔT boiler pinch point (°F)	30
Supplemental firing	0
Heat Rejection	Wet cooling tower

PERFORMANCE AND COST

Thermodynamic efficiency (percent)	--
Powerplant efficiency (percent)	35.5
Overall energy efficiency (percent)	35.5
Plant capital cost ($ x 10^6)	378
Plant capital cost ($/kWe)	436
Cost of electricity (mills/kWh)	25.2

NATURAL RESOURCES

Coal (lb/kWh)	0.88
Water (gal/kWh)	
Total	0.64
Cooling	0.44
Processing	0.08
Makeup	0.12
NO_x suppression	0
Stack gas cleanup	0
Land (acres/100 MWe)	8.98

ENVIRONMENTAL INTRUSION

	Lb/10^6-Btu Input	Btu/kWh	Lb/kWh Output
Gas Turbines			
SO_2	0.05		0.47×10^{-3}
NO_x	0.20		1.90×10^{-3}
HC	0		0
CO	0		0
Particulates	0		0
Gasifier System			
SO_2	0.20		1.88×10^{-3}
NO_x	0		0
Heat to water		1562	
Heat, total rejected		6199	

	Lb/kWh	Lb/Day
Wastes		
Elemental sulfur	33.4×10^{-3}	0.702×10^6
Ash (wet)	99.3×10^{-3}	2.087×10^6

MAJOR COMPONENT CHARACTERISTICS

Major Component	Unit or Module: Size (ft) (W x L (or D) x H)	Weight (lb) (x 10^6)	Cost ($ x 10^6)	Units Required	Total Cost ($ x 10^6)	$/kW Output
Gas turbine-compressor-combustor-generator	15 x 70 x 21	0.92	13.8	3	41.4	47.3
Heat recovery steam generator	48 x 39 x 64	0.83	4.9	3	14.7	16.8
Steam turbine-generator	120 x 12 x 14	1.24	10.3	1	10.3	11.8
Gasifier system (including booster steam-turbine-compressor)	--	--	--	--	93.0	106.2

Table 2.3-5

SENSITIVITY EFFECTS FOR WATER-COOLED GAS TURBINE

Variations from Base Case	Effect of Variations (mills/kWh)
Dry cooling towers	Adds 0.3
Ceramic nozzle	Reduces 0.7
Pressure ratio of 12	Reduces 0.6
3000 F firing	Reduces 1.3
Steam bottoming conditions:	
1800 psi	Reduces 0.2
1800 psi/1000 F/1000 F	Reduces 0.4

For the water-cooled turbine the utility of the heat transferred for cooling is far less. This thermal energy becomes useful to produce power only in proportion to the small fraction made into steam in the flash tank. Although the water is a powerful coolant, its thermal energy increase is not efficiently convertible to useful power.

The influence of alternate fuels is indicated in Table 2.3-6, with solvent refined coal showing the lowest electricity production cost of all cases studied.

Table 2.3-6

SENSITIVITY TO FUELS AT 2800 F, 16 PRESSURE RATIO FOR WATER-COOLED GAS TURBINE

Fuel	Output (MW)	Cost	
		$/kW	Mills/kWh
Low-Btu (base)	865	436	25.2
High-Btu	762	235	29.4
Solvent refined coal	728	266	23.6
COED	735	262	28.3

Table 2.3-7 presents the makeup of power generation and of plant auxiliary power consumption that resulted in the net plant output for each of the 28 cases.

The low bucket and nozzle temperatures realized with water cooling, of the order of 900 F (755 K), may permit exploitation of more corrosive fuels such as SRC. There are no small passages

Table 2.3-7

POWER OUTPUT AND AUXILIARY POWER DEMAND FOR BASE CASE AND PARAMETRIC VARIATIONS: OPEN-CYCLE GAS TURBINE COMBINED CYCLE—WATER COOLED

CASE NO.		1	2	3	4	5	6	7	8	9	10
PRIME CYCLE POWER OUTPUT	MW	690.3	724.2	714.6	555.0	522.6	527.1	460.2	920.4	795.0	646.5
BOTTOMING CYCLE POWER OUTPUT	MW	210.6	203.5	204.3	233.7	231.5	233.7	140.9	283.4	240.4	223.8
FURNACE POWER OUTPUT	MW	0.	0.	0.	0.	0.	0.	0.	0.	0.	0.
BALANCE OF PLANT AUX. POWER REQ'D.	MW	22.0	22.0	21.9	22.3	22.3	22.3	12.0	31.0	22.5	22.1
FURNACE AUX. POWER REQ'D.	MW	9.0	8.3	6.4	0.	0.	0.	6.0	12.0	10.4	8.9
TRANSFORMER LOSSES	MW	4.5	4.6	4.6	3.9	3.8	3.8	3.0	6.0	5.2	4.4
NET STATION OUTPUT	MW	865.4	892.7	886.0	762.5	728.0	734.7	580.1	1154.8	997.3	834.9

CASE NO.		11	12	13	14	15	16	17	20	23	24
PRIME CYCLE POWER OUTPUT	MW	677.1	690.3	756.3	688.8	697.8	683.1	690.3	690.3	690.3	690.3
BOTTOMING CYCLE POWER OUTPUT	MW	202.3	207.9	260.8	204.1	207.0	212.0	208.6	215.9	210.6	195.0
FURNACE POWER OUTPUT	MW	0.	0.	0.	0.	0.	0.	0.	0.	0.	0.
BALANCE OF PLANT AUX. POWER REQ'D.	MW	22.3	21.7	22.5	21.3	21.9	22.0	22.1	22.0	22.8	22.0
FURNACE AUX. POWER REQ'D.	MW	8.5	9.0	10.4	8.5	9.0	9.0	9.0	9.0	9.0	9.0
TRANSFORMER LOSSES	MW	4.4	4.5	5.1	4.5	4.5	4.5	4.5	4.5	4.5	4.4
NET STATION OUTPUT	MW	844.2	863.0	979.1	858.6	869.4	859.6	863.3	870.7	864.6	849.9

CASE NO.		25	26	27	28
PRIME CYCLE POWER OUTPUT	MW	690.3	690.3	690.3	690.3
BOTTOMING CYCLE POWER OUTPUT	MW	209.3	207.3	214.3	216.0
FURNACE POWER OUTPUT	MW	0.	0.	0.	0.
BALANCE OF PLANT AUX. POWER REQ'D.	MW	22.1	22.1	21.7	21.8
FURNACE AUX. POWER REQ'D.	MW	9.0	9.0	9.0	9.0
TRANSFORMER LOSSES	MW	4.5	4.5	4.5	4.5
NET STATION OUTPUT	MW	864.0	862.1	869.4	871.0

to be clogged by combustion products such as would be found on advanced air-cooled gas turbines. In addition the stoichiometric firing temperature for firing SRC would be greater than that for the low-Btu fuels considered. Higher firing temperatures could be developed burning SRC.

A combination of the best parametric conditions has been estimated although the point was not calculated. A coal pile to plant net output efficiency of 40 percent should result using ceramic stationary parts in a 3000 F (1922 K), 16 pressure ratio gas turbine low-Btu gasifier plant with steam bottoming conditions of 2400 psig, 1000 F/1000 F (16.7 MN/m^2, 811 K/811 K).

OBSERVATIONS

The water-cooled gas turbine combined power plant would provide high efficiency and may be highly tolerant of poor fuel quality. The high specific output of the gas turbine would reduce the number of turbines required and the balance of plant costs. The result is a low installed cost in dollars per kilowatt and a low cost of electricity production.

RECOMMENDED CASE

Parametric Case 9, with three gas turbines at 3000 F (1922 K), 16 pressure ratio and steam turbine conditions of 1450 psig, 1000 F (10.1 MN/m^2, 811 K) is recommended for further consideration. Alternative steam conditions with reheat should be considered. As an alternative to the low-Btu gas fuel, the use of solvent refined coal merits consideration.

Appendix A

GROUNDRULES FOR STUDY

Groundrules for the study were established. These guidelines were utilized by all subcontractors.

TYPE OF LOAD

Emphasis was on baseload central station plants. A baseload plant is defined as a plant operating at high capacity factors, as defined under "Power Output" below.

Although the term baseload does not imply wide or rapid load variations in normal operation, it is impossible to avoid infrequent conversion system or electric power system disturbances which result in the sudden loss of all or part of the plant load. The plant should be self-protecting in this event, and, preferably, the control should permit continuation of service at a reduced local load at rated voltage and frequency if the fault lies in the electric power system.

Intermediate load and peaking capability will be investigated for the energy conversion systems in Task III. This investigation included identification of technical factors that inherently limit load following and part load operation.

POWER OUTPUT

A range of rated capacity from 24 to 2400 MW was evaluated in the study; within these limits, each energy conversion system had a specified capacity range that was defined for Task I in the final matrix of point variations (included in Volume I).

Plant rated capacity is defined as the continuous electrical power output from the transmission voltage side of the transformer, with rated system pressures, temperatures, and flow rates, and normal makeup of the working fluid. Rated capacity is the net power, after subtracting all auxiliary and station service power. The power for all auxiliaries (pumps, compressors, vacuum pumps, etc.), whether electrically or mechanically driven, must be accounted for.

For each conversion system, an availability of at least 90 percent was assumed, and a capacity factor, defined as

$$\frac{\text{Kilowatt hours generated per year}}{\text{(8760 hours per year) (rated capacity in kilowatts)}}$$

of 65 percent was assumed for economic analysis. (This is equivalent to 5694 hours per year at rated capacity.)

The rated capacity is the output when new or refurbished.

Power was produced at 60 Hz, three phase, at transmission line voltage at the output side of the transformer. For those systems where size permitted direct connection to a distribution system, power was produced at 60 Hz at a distribution voltage.

EMISSION STANDARDS

Emissions standards used in the contract for each conversion system and each fuel were specified. In Task I, the emissions constituents of interest were SO_2, NO_X, CO, unburned hydrocarbons, and particulate matter. Emission control equipment, considered in performance and cost analyses, was utilized to maintain emissions below the specified amounts.

EXTENT OF THE PLANT TO BE STUDIED

For both performance and cost analyses, the extent of the plant accounted for is defined below.

For those systems using direct coal combustion, the following subsystems were included in the study:

- Coal handling equipment at the central station plant, including facilities for coal unloading from rail cars, storage facilities for a 60-day supply, and conveyor equipment (but not including coal transportation from the mine to the plant site)
- Combustors and emission control equipment
- Ash and other waste removal and disposal equipment within the plant site property limits
- The energy conversion systems, including heat input heat exchangers, and electrical generators
- All auxiliaries and balance of plant, including buildings, land, offices, shop facilities, special maintenance equipment, water treatment equipment, protective devices, etc.
- Power and voltage control subsystems suitable for isolated operation or operation in parallel with existing generating units in a utility system
- Heat rejecting subsystem
- Current inversion equipment, for those conversion systems generating direct current
- Transformers to raise voltage to transmission line or distribution voltage, but not the high voltage breaker and switch yard

For those systems using liquid, or intermediate- or high-Btu gas, as a fuel, it was assumed that the fuel will be generated at a remote facility and delivered to the power plant by

pipeline or tanker. The analysis did not include the liquefaction or gasification plant and transportation system, but did include a 60-day liquid fuel storage facility if applicable and all the applicable subsystems listed above except the coal handling equipment.

If enriched air, hydrogen, or oxygen supply streams were required for any of the energy conversion concepts under consideration, such streams were assumed to be available at the plant site property limits. All the applicable subsystems listed above were also analyzed.

For most conversion systems studied, at least one configuration using low-Btu gas was investigated in Task I. For cases where the gasifier and the conversion system were integrated, the study included the gasifier in addition to those applicable subsystems listed above. Common Study Teams conducted the integrated performance evaluation and supplied cost data for the conversion system components.

UNIT RATING AND SIZING

Units were rated at an average daily ambient temperature for the Middletown, U.S.A. site, 59 F (288 K) with a coincident relative humidity of 60 percent. Ambient air absolute pressure was 1 atm (101.3 kN/m^2). For process and makeup water, a water inlet temperature of 57 F (287 K) was used.

The generator and fuel handling capacity of the plant was sized and costed to handle a maximum power output which was expected at low-temperature operation, with ambient air at 20 F (267 K), a coincident relative humidity of 60 percent, and an ambient air absolute pressure of 1 atm (101.3 kN/m^2). The corresponding inlet water temperature was 39 F (277 K).

The heat rejection equipment was sized and costed for the 5 percent summer condition (as defined under "Heat Rejection Conditions" below).

HEAT REJECTION CONDITIONS

For heat rejection to air, available air inlet temperature of 94 F (308 K) was used with no specified maximum rise. Ambient air absolute pressure was 1.00 atm (101.3 kN/m^2).

For heat rejection in an evaporative cooling tower, available inlet air wet bulb temperature 76 F (298 K) was used. Ambient air absolute pressure was 1 atm (101.3 kN/m^2).

Evaporative cooling towers were established as the base case heat rejection method.

BASIS FOR CAPITAL COSTS

Component capital costs were broken down and presented by the advocates in terms of the following cost elements:

- Material costs expressed in 1974 dollars
- Labor man-hours by labor category

Labor rates and escalation adjustments were applied by the architect-engineer, using the craft rates for the Middletown, U.S.A. site, updated to the first half of 1974 as specified by NASA.

In calculating the annual cost of the capital investment, a fixed charge rate of 18 percent (to be modified as appropriate during the contract) was applied to the total capital cost for each of the conversion systems used. The fixed charge rate was assumed to include:

- Interest return to the bondholders
- Equity return to the stockholders
- Federal and State income taxes
- Depreciation (based on a 30-year useful life for all nonexpendable plant components)
- Local property taxes
- Insurance

The amortization of research and development costs was not included in the annual cost.

BASIS FOR OPERATING COSTS

As indicated under "Power Output," a capacity factor of 65 percent was assumed as the base value for all conversion systems being studied.

Cost of coal- or coal-derived clean fuel was specified for use in the study. Fuel costs* include transporting the fuel to the power plant site.

Advocates proposed schedules of estimated inspections and maintenance required, including down time, labor, and materials. These estimates were assessed and subsequently modified in deriving operating cost estimates.

EFFICIENCY AND HEAT RATE

In presenting data on heat rates, cost of fuel heating value, etc., the higher heating value of the fuel was used at all times for consistency.

Efficiencies and heat rates were net, based on the higher heating value of fuel as received and on rated electrical capacity as defined under "Power Output."

*Fuel costs were specified.

SITE LOCATION

It was assumed that the site location is "Middletown, U.S.A.," as defined in Guide for Economic Evaluation of Nuclear Reactor Plant Designs, USAEC Report NUS-531, Appendix A, January 1969, as modified by the ambient air and water temperatures specified under "Unit Rating and Sizing," and "Heat Rejection Conditions." Certain aspects of the location, such as the nominal distance from a coal mine were selected.

COAL TYPES

Three typical types of coal were investigated. These coals were:

- Illinois No. 6
- Montana sub-bituminous
- North Dakota lignite

STACK GAS TEMPERATURE

For all conversion systems analyzed, a stack gas temperature no lower than 250 F (394 K) was used under all operating conditions.

Appendix B

COST AND PERFORMANCE EVALUATION COMPUTER PROGRAM

INTRODUCTION

In order to process the large quantity of data and to ensure that the performance and cost of the ten candidate systems would be calculated in a consistent manner, a basic computer program was written first. This program was then converted to twelve programs, with each one being modified as needed to tailor it to a specific system or configuration. Only a few systems, as will be discussed later, required modifications to the performance calculations. All programs required some changes in the cost section, however, since different major components are needed by each. The computer printouts for the specific energy conversion systems are integrated with the cycle sections in Volume II.

The cost data were input in millions of dollars per unit followed by the number of such units required for all major components as well as cooling tower cost, site labor cost, and all other balance-of-plant cost.

Two types of output were obtained from the programs. The first was performance data, which included such items as various types of efficiencies, coal consumption, and cost of electricity. The second was a tabulation of capital cost distribution, which included, in millions of dollars and in dollars per kilowatt, the cost of all major components, all balance-of-plant cost, contingency, escalation costs, and interest during construction.

A brief description of the various performance parameters and capital cost distribution figures evaluated in the basic program is given below.

PERFORMANCE DATA CALCULATED AND PRINTED BY BASIC PROGRAM

Actual Power Plant Output. This is the actual megawatts of power which will be put into the transmission line leaving the power plant. It is obtained by subtracting the auxiliary power, if any, used by the furnace and that used for the balance of plant from the net power output of the prime cycle and bottoming cycle, if any, and furnace, if such power is produced. These new power outputs (one, two, or three, depending on the system) were obtained by allowing for the inefficiency associated with converting the power to the required transmission voltage.

Thermodynamic Efficiency. This efficiency is calculated by dividing the gross power output of the prime cycle and bottoming cycle, if any, by the heat delivered to the prime cycle and to the bottoming cycle, where such additional heat is used.

Power Plant Efficiency. This efficiency is calculated by dividing the actual power plant output (as defined earlier) by the higher heating value of the coal or clean fuel used.

Overall Energy Efficiency. This efficiency, which is also referred to as the "coal pile to bus bar efficiency " is calculated by dividing the actual power plant output by the higher heating value of the coal used, i.e., the higher heating value of the coal used when direct combustion of coal is employed and the higher heating value of the coal used to produce the required clean fuel when clean fuel is employed. The overall energy efficiency, the same as the power plant efficiency for cases using direct combustion of coal and for cases using clean fuel, is lower than the power plant efficiency by a factor which is equal to the conversion efficiency of the clean fuel producing process.

Coal Consumption. The coal consumption, on a per kWh basis, is based on the total coal used. In the case of clean fuel systems, it is the coal used to produce the required clean fuel.

Plant Capital Cost. This cost figure, in millions of dollars, is the total plant capital cost and includes the cost of all system components, all balance-of-plant cost, contingency, escalation costs, and interest during construction. A more detailed breakdown of capital cost distribution is given in a second tabulation and will be discussed later.

Cost of Electricity. On the basis of a 0.65 capacity factor, the cost breakdown of electricity in mills/kWh is calculated for 1) capital, 2) fuel, and 3) operating and maintenance.

Capital. The cost of electricity due to capital costs is calculated on the basis of an 18 percent fixed charge rate. This rate, which was supplied by NASA, is made up as follows:

7.5 percent—cost of money
4.1 percent—F.I.T.
3.3 percent—depreciation (30 years)
2.8 percent—other taxes
0.1 percent—insurance
0.2 percent—working capital

The electricity cost due to capital cost was, therefore, calculated simply by taking 18 percent of the total capital cost, converting it into mills and dividing by the kWh of power output per year based on operating 65 percent of the time.

Fuel. The cost of electricity due to fuel charges was obtained by dividing the hourly fuel costs in mills by the actual power output in kW.

Operating and Maintenance (O&M). The cost of electricity due to maintenance and operating expense was obtained by dividing the total yearly maintenance and operating costs by the kWh of power output per year based on operating 65 percent of the time.

Total. This value is the total cost of electricity and is the sum of the above three.

Sensitivity. To determine the sensitivity of the cost of electricity to various cost factors, four additional cost figures were determined. The first two of these are the total cost of electricity, as defined above, based on a capacity factor of 0.50 and 0.80, respectively. The third and fourth are the change in the cost of electricity for a 20 percent change in the cost of capital and fuel, respectively.

CAPITAL COST DISTRIBUTION CALCULATED AND PRINTED BY BASIC PROGRAM

Major Components. The cost of all major components, in millions of dollars, is printed out in the categories of 1) prime cycle, 2) bottoming cycle, and 3) primary heat input and fuel system. The subtotal of all major components is then given.

Balance of Plant. The balance-of-plant cost, also in millions of dollars, is broken down into the categories of 1) cooling tower, 2) all other component cost, and 3) site labor. This is also followed by the sub-total value.

Contingency. The contingency figure is a straight 20 percent of the total of all major component costs and balance-of-plant costs.

Escalation Costs. The added cost due to escalation was computed on the basis of an escalation rate of 0.065 per year. The cumulative cash flow during power plant design and construction was assumed to follow the S-shaped curve for fossil fuel plants shown in Figure 9 of Reference 1. The abscissa of this curve was converted to the dimensionless parameter n/N, where n is number of years from start of design and N is the total number of years for design and construction. In this way the polynomial curve fit of the resulting curve was independent of total design and construction time and could therefore be used for all systems. The conservative assumption was made that for each year the escalation cost would be based on the escalation existing at the end of the year. Thus, for the nth year the escalative cost is:

$$C_n (1 + e)^n - C_n$$

where C_n = unescalated cash flow during the nth year and e = escalation rate of 0.065. The total escalation cost therefore becomes:

$$\sum_{n=1}^{N} C_n (1 + e)^n - C$$

where N = total years for design and construction and C = total unescalated cash flow.

Interest During Construction. The added cost due to interest during construction was computed on the basis of an interest rate of 0.10 per year. The cumulative cash flow during power plant design and construction was again assumed to be the same S-shaped curve used for the calculation of escalation cost but with the computed escalation cost added on. Thus, for the nth year the interest cost is:

$$[C_n \ (1 + e)^n] \ [(1 + \frac{i}{2}) \ (1 + i)^{N-n} - 1]$$

where i = interest rate of 0.10 and all other symbols have the same meaning as in the discussion of escalation costs. The total interest cost therefore becomes:

$$\sum_{n=1}^{N} [C_n \ (1 + e)^n] \ [(1 + \frac{i}{2}) \ (1 + i)^{N-n} - 1]$$

Total Capital Cost. The total capital cost, in millions of dollars, is then just the sum of the major component costs, balance-of-plant costs, contingency, escalation and interest (therefore no longer in mid-1974 dollars but dependent on time for construction).

In addition to the above capital cost distribution figures, the major costs were also computed and listed in terms of dollars per kilowatt.

PROGRAM DEVIATIONS

As mentioned earlier, the performance calculations of the basic computer program had to be modified slightly for some systems because of peculiarities associated with these systems. Such modification will be discussed here.

The thermodynamic efficiency was not calculated for the fuel cell cycles nor for those open-cycle gas turbine-combined cycle cases in which an integrated low-Btu (LBtu) gas system was used.

In the case of the open-cycle MHD systems where LBtu gas is required for seed reprocessing, the higher heating value of this LBtu gas was included with the heat to the prime cycle in the calculation of the thermodynamic efficiency. In the calculation of the overall energy efficiency, the higher heating value of the coal required to produce this LBtu gas was added to the higher heating value (HHV) of the coal used by the prime cycle (either in direct combustion or in the production of the clean fuel). The cost of the seed reprocessing LBtu gas was added to the prime cycle fuel cost when calculating both the cost of electricity due to fuel cost and total cost of electricity. One open-cycle MHD system also required the use of oxygen. The cost of this oxygen, considered as an over-the-fence product, was also added to the fuel costs when calculating cost of electricity; however, it was considered to have a fixed price per ton and was, therefore,

not included when calculating the change in cost of electricity due to a change in fuel charges.

The low-temperature fuel cell systems had a number of peculiarities which required changes in the performance calculations

1. Oxygen is required in one of the low-temperature fuel cell cases. The cost of this oxygen was added to the fuel cost when calculating cost of electricity, but, as was the case with the open-cycle MHD system, the price of the oxygen was considered to be fixed and was therefore not included in the calculation of change in cost of electricity due to a change in fuel charges.

2. In the operation of the low-temperature fuel cells, there is fuel (mostly in the form of methane) remaining in the fuel stream after passing through the fuel cells. This residual fuel is assumed to be returned to the fuel-processing plant for a cost credit. This cost credit was input to the program as a dollars/hour amount and was subtracted from the original fuel costs.

3. At prescribed intervals, it is necessary to reprocess the platinum and replace the electrolyte in the low-temperature fuel cells. The cost of such refurbishing is included in the final operating and maintenance costs. The inputs required by the program to calculate these refurbishing costs are: 1) total electrolyte area; 2) labor costs per square foot of electrolyte area; 3) platinum loading in grams per square foot of electrolyte area; 4) dollars per grain of platinum reprocessed and; 5) number of hours between reprocessing. With these inputs the computer program calculated the refurbishing cost on a dollars per hour basis and added this value to the hourly O&M cost figure.

4. For all other systems, the O&M changes were supplied as input to the computer program. These changes are presented in Appendix C of Volume II, Part 1.

REFERENCE

1. Power Plant Capital Costs: Current Trends and Sensitivity to Economic Parameters, Report WASH-1345, U.S. Atomic Energy Commission Contract W-7405-eng-26, Oak Ridge National Laboratory; Contract AT (11-1)-2226, United Engineers and Constructors Inc., U.S. Government Printing Office, October 1974, 74 pp.

Appendix C

OPERATING AND MAINTENANCE COST ESTIMATES

INTRODUCTION

In February 1973, the Federal Power Commission (FPC) reported that in 1971 fuel costs for fossil-fueled steam electric plants accounted for 80 percent of the total electrical energy production costs, and that plant operating and maintenance (O&M) costs accounted for the remaining 20 percent (ref. 1). The O&M costs included plant labor for operations, supervision and engineering, and maintenance, as well as plant operating supplies, including lubricants, chemicals, other miscellaneous materials, office and other incidental expenses, and maintenance renewal parts and materials. The total 20 percent was divided about equally between operation and maintenance functions.

An FPC report on gas turbine plants states that in 1972 fuel costs accounted for 83 percent of the total electrical energy production expenses for major gas turbine installations (ref. 2). Of the remaining 17 percent, operation costs accounted for 17 percent and maintenance activities for 83 percent. Maintenance requirements varied widely among different gas turbine plants, depending on such factors as hours of operation, frequency of startup, mode of operation, and fuel quality.

Because of the limited time available in Task I, it was not possible to make detailed estimates of O&M costs for the wide variety of advanced energy conversion cycles and heat sources. It was decided to draw on the FPC data to select some base values for O&M costs and to adjust those values as deemed necessary by unusual operating conditions in a particular cycle. The following discussion describes how the base values were selected, adjusted, and used as inputs to the computer program for calculating the estimated cost of electricity.

SELECTION OF BASE VALUES FOR O&M COSTS

In Reference 1, the O&M costs for fossil-fueled steam electric plants covered a range from 0.6 to 4.9 mills/kWh. The weighted average for plants reported for 1971 was 0.94 mills/kWh; this cost was 13 percent higher than for 1970.

In Reference 2, it was found that the O&M costs for the open-cycle gas turbine plants reported for 1972 (first annual publication) ranged from 0.9 to 4.4 mills/kWh. The average capacity factor for these plants was 14 percent.

After studying the FPC data and holding discussions with General Electric specialists in the area of turbine plant maintenance, it was decided to assume the following parameters as base values for O&M costs:

Open-Cycle Gas Turbines	1.6 mills/kWh
For turbine inlet temperature less than 2200 F and burning "clean fuels"	
Advanced Steam Cycle	2.0 mills/kWh
For turbine inlet temperatures of 1000 F	

Using the base values of 1.6 mills/kWh and 2.0 mills/kWh for the gas and steam turbines, respectively, methods were established to estimate O&M costs for unusual operating conditions in advanced cycles. The assumptions and application of these methods are described in the following section.

ASSUMPTIONS AND CALCULATION METHODS FOR O&M COSTS

Except for open-cycle gas turbines and steam cycles, there is no historical data base which can be used to assume values for O&M costs. The capital cost estimates developed in Task I do, however, provide a basis for estimating increments of maintenance cost. Therefore a logic was developed to add increments of maintenance cost to assumed base values in order to account for the maintenance of components which have no precedent in real operational experience. If a component is believed to require maintenance cost which is not accounted for in the base values, an annual amount is added to the base O&M cost. This annual amount is determined by an assumed percentage of the original capital cost of the component.

Tables C-1 and C-2 summarize the O&M base values and component maintenance adders for the various cycles.

TABLE C-1

OPERATING AND MAINTENANCE COST ASSUMPTIONS

Cycle	O&M Base Value (mills/kWh)	Components Requiring Additional Maintenance Cost (where applicable)
Open-Cycle Gas Turbine		
Turbine Inlet T<2200 F, "Clean Fuel"	1.6	
Turbine Inlet T≧2200 F, "Clean Fuel"	2.0	
Turbine Inlet T≧2200 F, "Semi-Cl. Fuel"	2.4	
Open-Cycle Gas Turbine –Combined Cycle	(same as above)	Low-Btu gasifier
Closed-Cycle Gas Turbine	1.8	Primary heat input system. Low-Btu gasifier
Supercritical CO_2 Cycle	1.2	Primary heat input system. Low-Btu gasifier
Advanced Steam Cycle		Primary heat input system. Low-Btu gasifier
Turbine Inlet T = 1000 F	2.0	
Turbine Inlet T = 1200 F	2.2	Emission Control Equipment
Turbine Reheat T = 1400 F	2.3	
Liquid Metal Topping Cycle	2.2	Primary heat input system. Low-Btu gasifier
MHD Cycles (all)	2.0	Primary heat input system. Low-Btu gasifier Emission Control Equipment. Special adders per Table C-2
Fuel Cells–Low Temperature	2.5	Special adders per Table C-2
Fuel Cells–High Temperature	2.0	Low-Btu gasifier Special adders per Table C-2

TABLE C-2

COMPONENT MAINTENANCE ADDERS

Component	Assumptions for Annual Adder as Percentage of Initial Capital Cost of Component	
Low-Btu Gasifier	6%	
Primary Heat Input System		
Atmospheric Fluidized Bed	4%	
Pressurized Fluidized Bed	4.5%	
Pressurized Furnace	2%	
Emission Control Equipment	3%	
Open-Cycle MHD	Coal	SRC
Combustor	20%	15%
Generator-Diffuser	20%	15%
Slagging Furnace	20%	15%
High-Temperature Air Preheaters	20%	15%
Closed-Cycle Inert Gas MHD		
Heat Input Heat Exchanger	15%	5%
Generator-Diffuser	10%	10%
Closed-Cycle Liquid Metal MHD		
Generator-Diffuser	10%	
Fuel Cells—Low Temperature		
Solid Polymer Electrolyte-Catalyst Renewal	($130,000/year)	
Phosphoric Acid Electrolyte-Catalyst Renewal	($330,000/year)	
Fuel Cells—High Temperature		
Fuel Cell Stacks	10%	

REFERENCES

1. Steam-Electric Plant Construction Cost and Annual Production Expenses, Twenty-Fourth Annual Supplement-1971, Federal Power Commission, Washington, D.C., February 1973.

2. Gas Turbine Electric Plant Construction Cost and Annual Production Expenses, First Annual Publication-1972, Report S-240, Federal Power Commission, Washington, D.C., July 1974.

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